INTERIM REPORT
INVESTIGATION OF POSITIVE-TYPE SHAFT SEALS

November 1966

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Contract NAS8-11325

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ROCKETDYNE • A DIVISION OF NORTH AMERICAN AVIATION, INC

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FOREWORD

Rocketdyne, a Division of North American Aviation, Inc., has prepared this report under National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Propulsion and Vehicle Engineering Laboratory, Huntsville, Alabama, Contract NAS8-11325, G.O. 8624. This report covers the period from July 1964 through December 1966.

ABSTRACT

A series of new type seal concepts were generated, and three of the most promising were detailed for fabrication and testing to evaluate the designs for future turbopump applications. Descriptions of the various concepts, basis for the final selections of the seals for evaluation, and results of testing are included.

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ILLUSTRATIONS

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INTRODUCTION

The continued advancement of rocket engine turbomachinery has required the development of new shaft seal technology to cope with the extremes of temperature and speed. Although seal engineering technology has progressed significantly, turbomachinery performance demands have also increased particularly in the areas of high speed, high pressure, throttleability (wide speed range), and extended life.

The static portion of the mechanical seal, referred to as the secondary seal, must maintain its integrity while accepting axial shaft displacements caused by elastic deformations, fluid pressure pulsations, and vibrations. This report summarizes efforts directed toward development of new concepts intended to solve some of the problems of secondary sealing.

At a meeting in Huntsville, Alabama, in August 1964, representatives of NASA and Rocketdyne finalized a program to investigate new approaches to secondary sealing methods. This program is an attempt to study new secondary seal concepts; pointing the way to achieving lower leakage, greater reliability, and longer life at operating conditions of greater speeds, higher pressures, and longer life of future turbopump designs.

SUMMARY

A program to investigate the secondary seal area of positive-type shaft seals was initiated in July 1964. As a result, a total of 18 seal concepts were evaluated; three of which were selected for fabrication and performance analysis.

The welded metal bellows was selected as the secondary seal for all three seals, each having a different method of controlling the oscillatory motion of the bellows caused by the mechanical vibration and pressure pulsating environment of the turbopump. Of the three concepts selected and tested, one design, known as the particle damped seal, has been shown to have the best damping characteristics.

Further refinement of the particle damped seal is recommended to improve the response of the seal utilizing metal particles to reduce the effect of displacement inputs.

SECONDARY SEAL PROGRAM

PROGRAM OBJECTIVES

The basic program was directed at generating new approaches to the problem of secondary sealing with emphasis on advancing seal technology for future turbopump generations.

To meet the program requirements the following tasks were established:

1. Generate a number of new approaches to the secondary seal based on advanced turbopump operating parameters
2. Evaluate these concepts to arrive at several of the most promising
3. Perform detailed analyses and designs resulting in procurement of test seals of the selected concepts
4. Conduct nonrotating tests to evaluate concepts and to provide information for future designs

DESIGN CONCEPT SELECTION

Seal design criteria were based on expected future turbopump performance requirements. Future pump discharge pressures are expected to be higher than current levels and the pressure at the seal cavity could readily be 100 percent above current practice. The seal temperature environment is expected to be in a range of -323 to +1000 F.

Seventeen seal concepts were evolved. After a thorough analysis and investigation, the following three designs were selected for detailed investigation and possible fabrication.

1. Piston Damped Seal (Fig. 1)
2. Purged Double Lip Seal (Appendix
3. Orifice Damped Seal (Fig. 2)

Upon further investigation, the purged double lip seal was dropped because of temperature limitations and susceptibility to contamination. Late in the program a new concept evolved, known as the particle damped seal and was the subject of effort under a program extension.

A description of the 17 seal, concepts and the results of the feasibility investigation, are included in the Appendix.

A result of the secondary seal selection evaluation described in the Appendix was the conclusion that the concept having the greatest number of advantages is the welded metal bellows. The metal bellows design provides the most positive method of preventing secondary seal leakage, with a minimum number of potential leak paths.

The high performance of the metal bellows mechanical seal can be attained if a means can be found for controlling the potentially unstable seal face movements induced by vibratory inputs and fluid pressure pulsations. The ability of the bellows to function as a stable secondary portion of the mechanical seal has a major part in controlling leakage and life of the sealing faces. As a result of the potentially unstable behavior of bellows, considerable attention has been given throughout the seal industry to the problems of bellows damping.

A conventional method of retarding unstable motion of bellows is by frictional devices, usually spring loaded, contacting the bellows convolutions and/or the bellows carrier. Although effective, the amount of damping is not easily controlled and damper material and contacting surfaces are subject to wear. In addition, if the input vibration becomes large, a higher frictional load is then necessary for adequate control. Simultaneously, this higher load increases energy input and the potential for ignition of exposed propellants. The three damping concepts selected for evaluation under the investigation covered in this report do not depend upon exposed friction damping.

Based on the selection of the metal welded bellows, a test program was outlined to study the effects of incorporating bellows vibration damping devices to ensure that the damper does not impair the normal operation of the bellows with reference to primary seal separation and bellows life. A test program was planned to include axial cycling of the bellows both mechanically and through pressure pulsations to observe bellows integrity and seal performance. A test was planned to accelerate the mating ring away from the bellows carrier to determine bellows response or recovery rate. Vibration tests were planned to observe the reaction of the bellows to vibration input. Other tests were also planned for analysis of total seal face loading and normal quality control inspection tests.

CONCLUSIONS

Piston Damped Seal

The piston damped seal (Fig. 1) employs a viscous method of controlling induced vibratory motion of the bellows and consists of a piston ring contained in the housing maintaining a close clearance with the carbon retainer

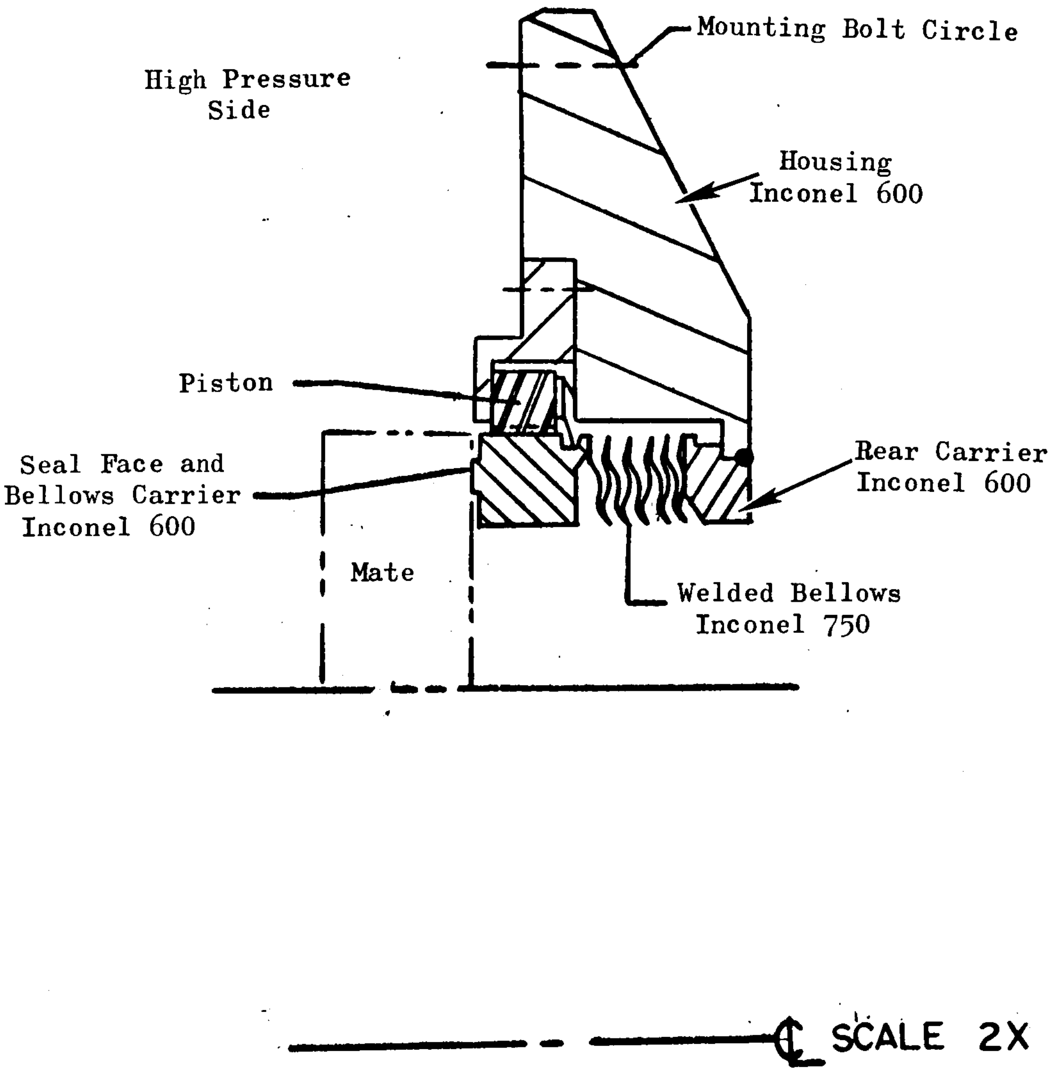

Figure 1 . Piston Damped Seal Design

outer diameter. The sealing fluid surrounding the bellows OD is forced through the controlled clearance as axial movement of the bellows changes the volume between the housing and bellows.

Based on initial computer studies describing damping characteristics in terms of displacement vs time response, the seal shows a definite reduction in amplitude when the piston clearance is reduced to 0.002 inches from 0.010 inches. A conclusion based on pressurized cryogenic tests conducted on actual hardware is that a relatively high density and viscosity is required. The tests also show that the use of a piston does not impair normal bellows operation, which indicates that excessive damping is not obtained under the selected test conditions. No apparent damping exists in a gas environment, which precludes the utility of this design in the turbine area.

Although damping is apparently obtainable in a cryogenic fluid when a close clearance is maintained, a further reduction in clearance is accompanied by rubbing and will approach the method of frictional damping currently employed in the seal industry. Because of a somewhat restricted use, the piston damped seal was ruled out for further refinement. A more practical use for this design would be in seals for oils or similar fluids of relatively high viscosity and for application where rubbing contact may be permissible.

Orifice Damped Seal

The orifice damped seal (Fig. 2) also utilizes viscous friction to absorb imposed axial vibration. The design consists of two cavities formed by two pairs of radially stacked welded bellows and separated by an orifice plate. The end fitting of one bellows cavity has a sealing surface to

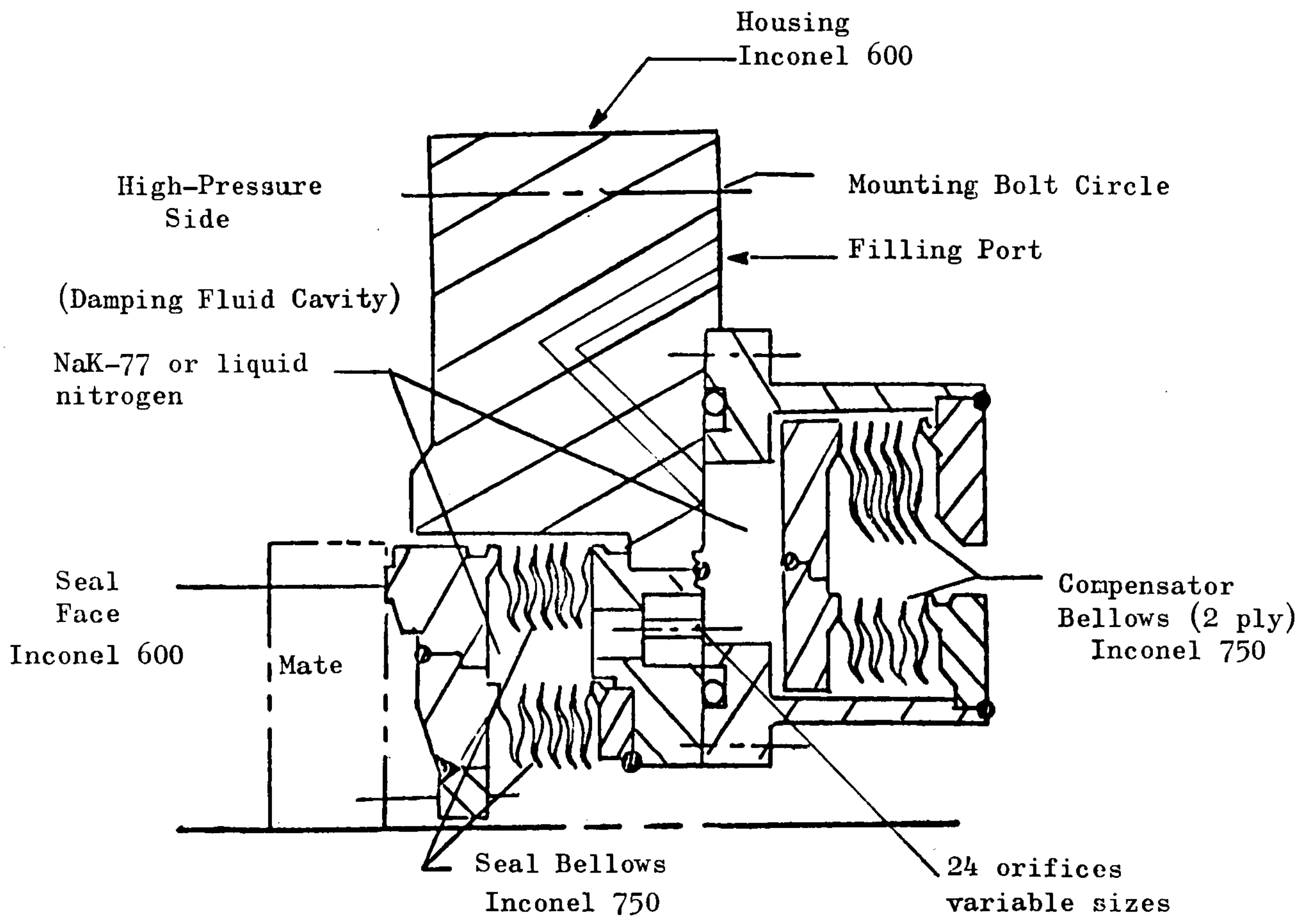

Figure 2. Orifice Damped Seal Design

are welded to a radial extension of the sealing surface placing the cylinders at the OD of the bellows. The cylinders are filled to an effective level with molybolenum spherical particles. The spherical particles react to vibration inputs by absorbing displacement energy through inertia and friction of the particle masses acting on the inside surface of the cylinders.

The prime advantages over conventional vibration damping devices and other concepts considered in this program are simplicity of design and reliability potential with no contamination or fire risk when in the proximity of propellants. In addition, effective damping can be obtained over a wide range of temperatures, from cryogenic to turbine gas environments.

Vibration data taken during testing of the particle damped seal indicate effective damping of seal nonrotating parts with the potential advantages of increasing carbon seal face life and improving leakage characteristics. Further analysis and testing is recommended to arrive at a configuration offering the most desirable damping characteristics and seal performance. Further discussion of the recommended test program is given in another section of the report.

CONCEPT DESIGN

PISTON DAMPED SEAL

Figure 4 depicts the final design of the piston damped seal. The parameters of this design are:

Mass (M) = 0.00065 lb sec^2/in.

Spring Rate (K) = 400 lb/in.

Bellows Effective Diameter = 3.60 inches

Piston Diameter = 3.80 inches

Radial Piston Clearances = 0.002, 0.004, 0.006, 0.008, 0.010 inch

The material of the welded bellows is Inconel 750 with 0.006-inch plate thickness. The piston is solid and has a radial clearance maintained constant by three small equally spaced pads on the piston, and contacting the bellows carrier OD.

During the parameter study involving a gas medium, no appreciable damping was obtained for this seal with the small axial motion involved. A damping coefficient of sufficient magnitude is obtained from a liquid nitrogen medium.

The generalized equation solved for use in the parameter study is

$$\frac{d^2x}{dt^2} + Kx + C\left(\frac{dx}{dt}\right)^2 = 0$$

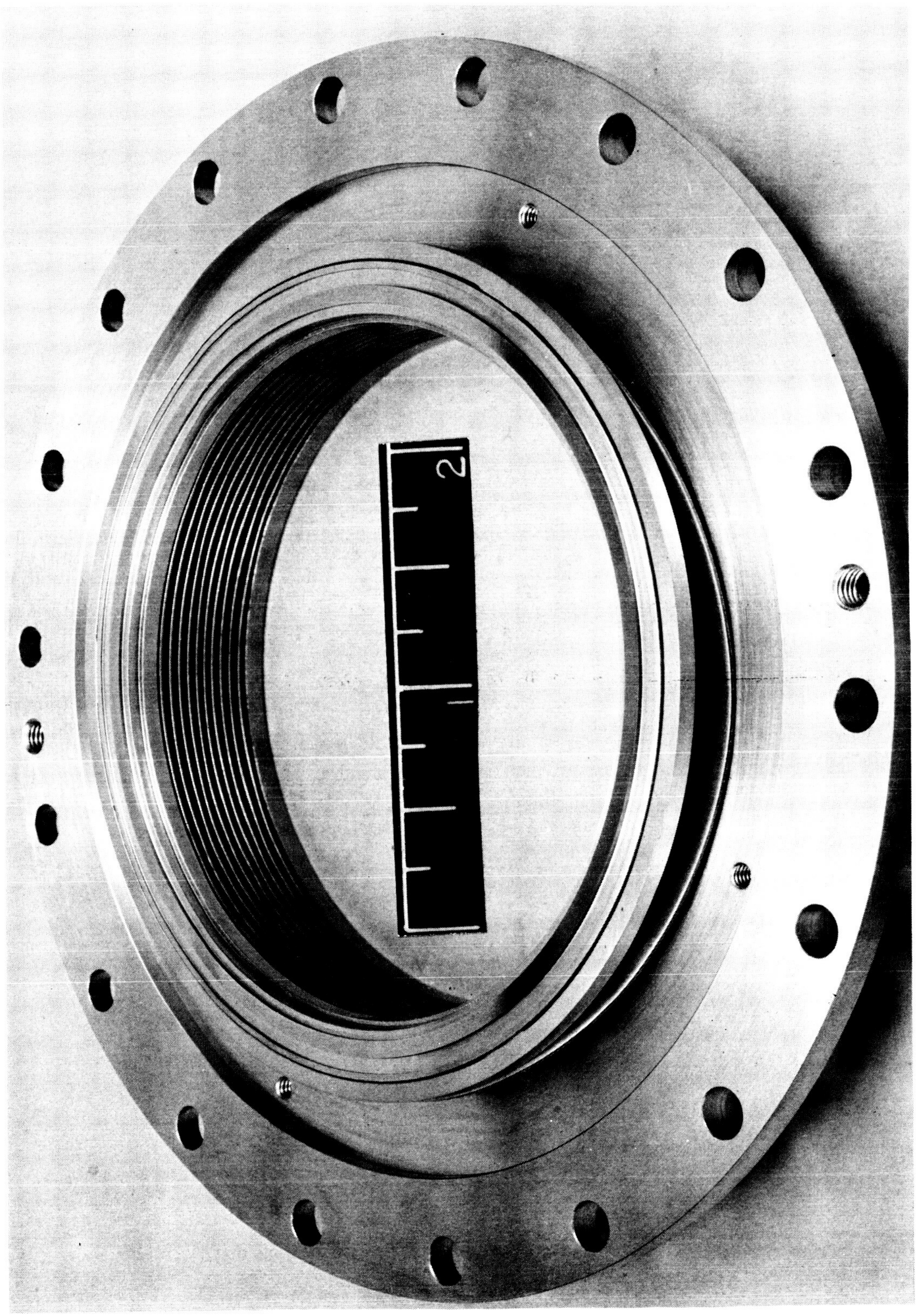

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Figure 4. Piston Damped Seal

Turbulent flow was assumed to exist through the piston ring clearance. Figures 5, 6, and 7 show the displacement vs normalized time for several piston ring clearances. Radial clearances from 0.002 to 0.010 inches were evaluated. Table 1 compares some of the results of the parameter study. The additional nomenclature used is as follows:

T = time lapse in milliseconds to reach zero displacement from a unit compression for turbulent flow through the piston clearance.

WN = natural frequency in cps of the spring mass system

C/C_c = Damping ratio of the system for turbulent flow through the clearance

The ratio C/C_c is the ratio of the damping coefficient (c) to the critical damping coefficient (C_c) of the system.

Upon concluding the parameter study which resulted in defining the physical size of the seal and establishing the limits of control parameters with respect to computed damping characteristics, the hydraulic balance and unit face load of the seal was considered.

Seal face loading is one of the primary factors affecting sealing and dynamic seal life, and is dependent on two forces; the spring load exerted by the bellows and the hydraulic force imposed on the bellows plates by the environmental pressure.

Because the bellows plates will deform due to either a change in pressure or deflected length (Δ l), the bellows characteristics are closely examined. The effect of bellows deformation changes the effective hydraulic area of the plates and directly affects the seal face loading. The hydraulic

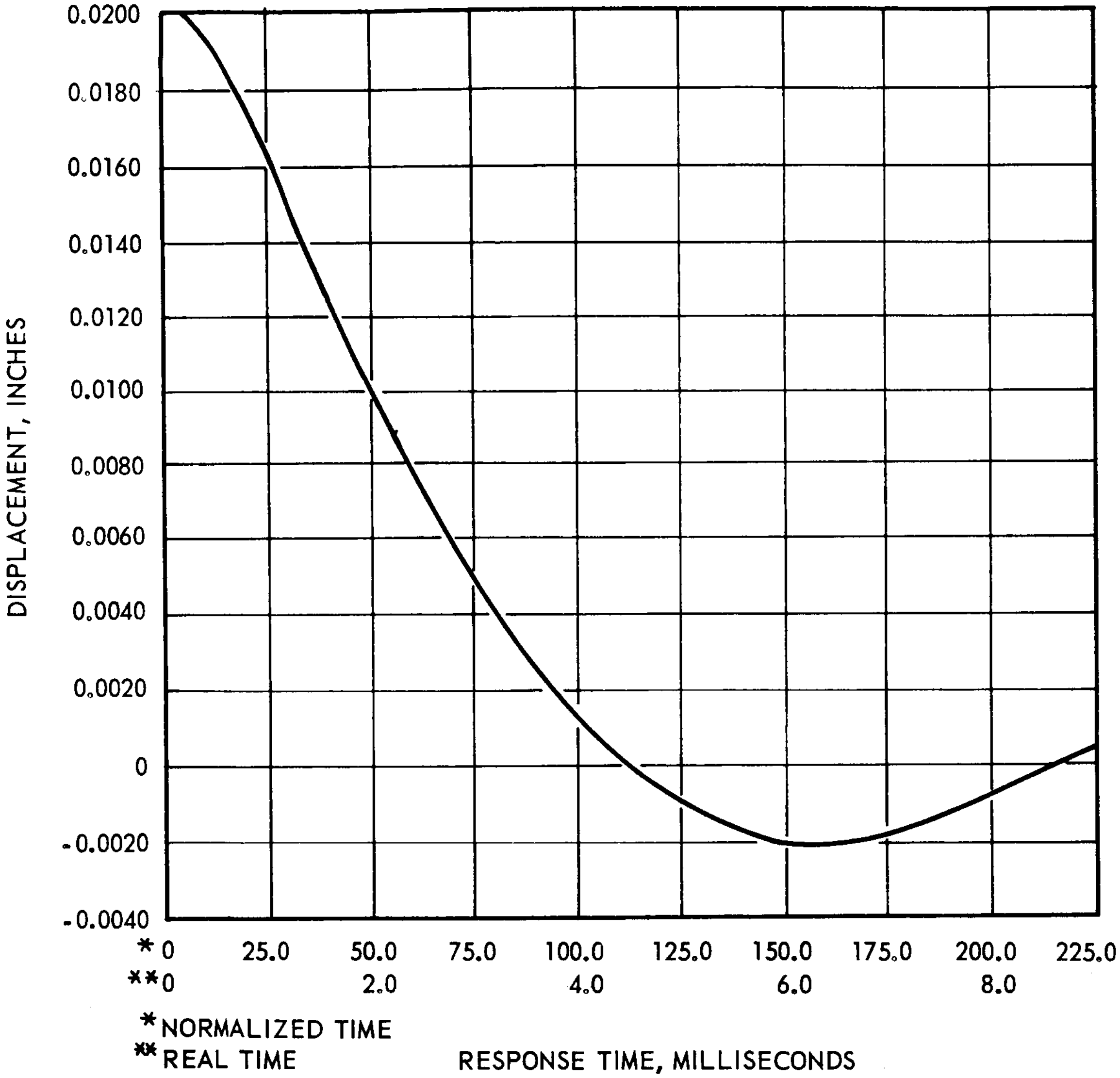


Figure 5. Computer Data on Piston Damped Seal, Radial Clearance = 0.002 inch

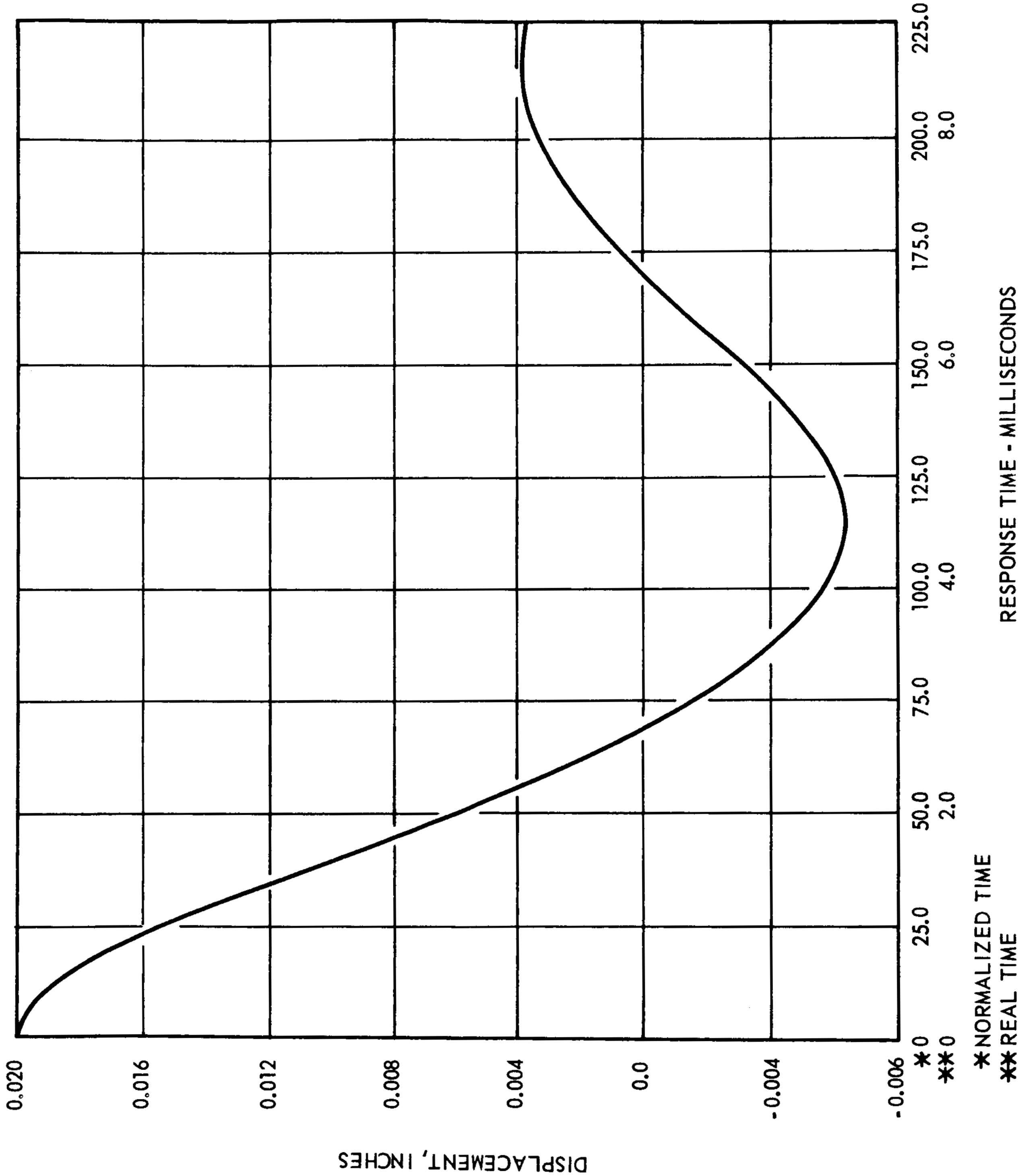


Figure 6. Computer Data on Piston Damped Seal, Radial Clearance = 0.003 inch

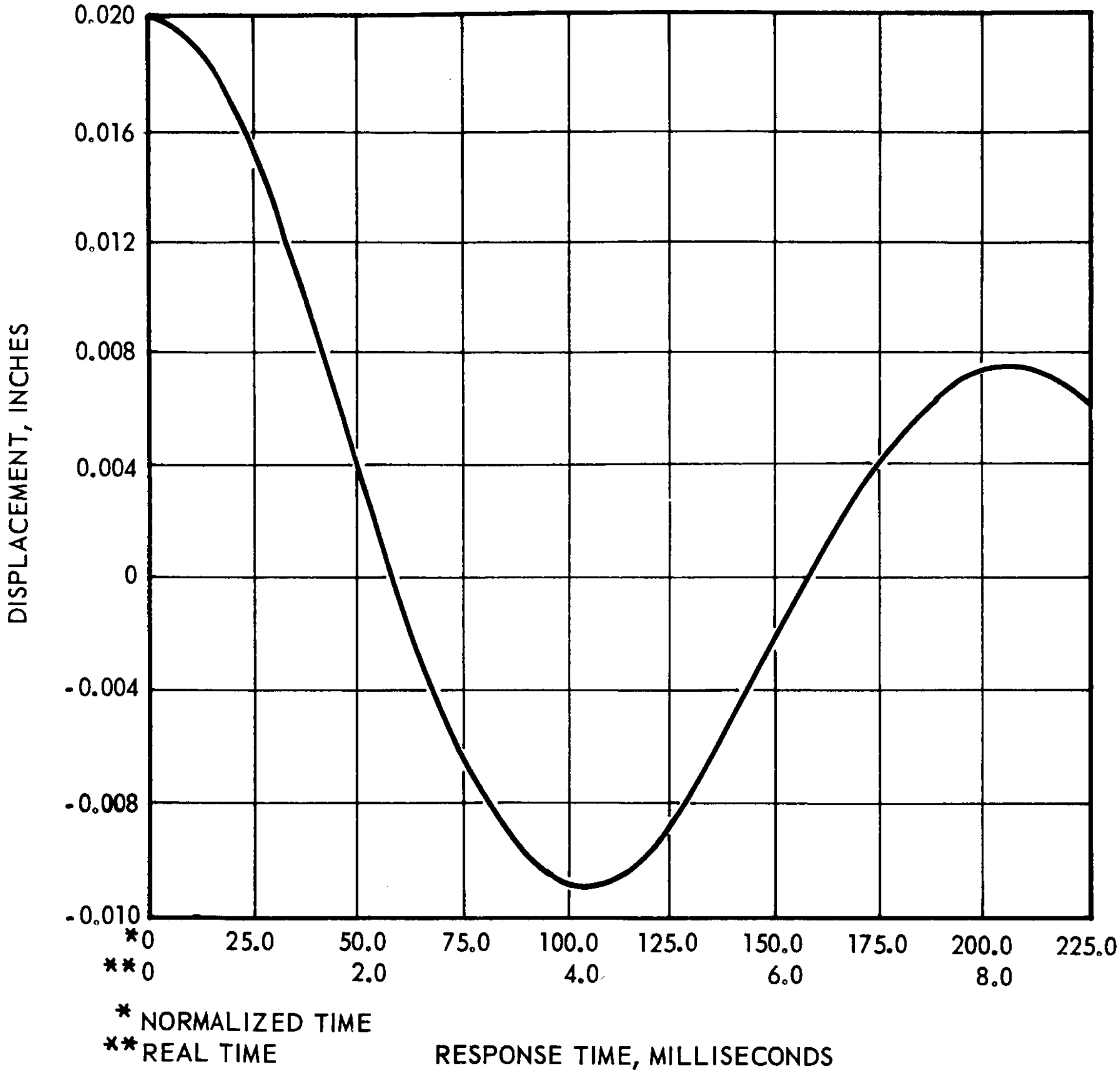
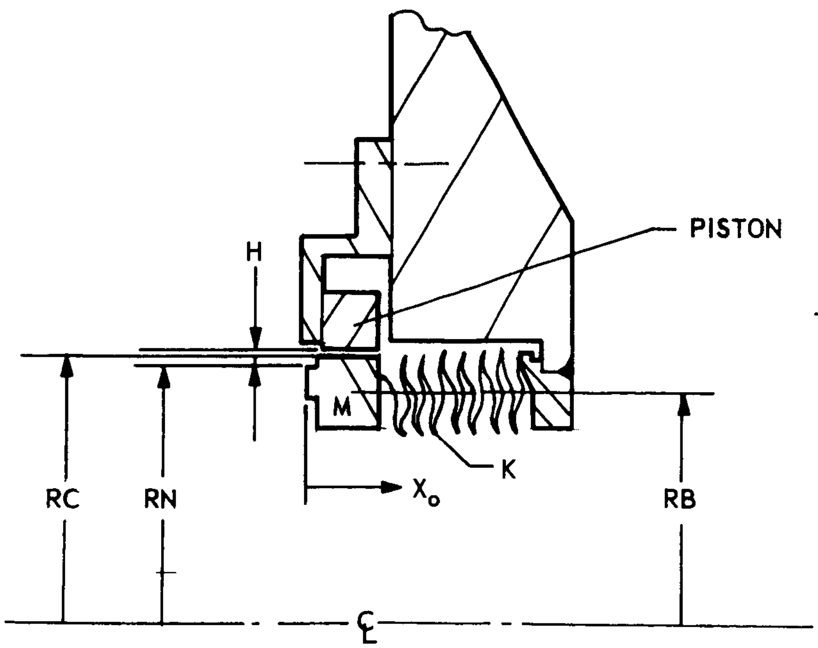


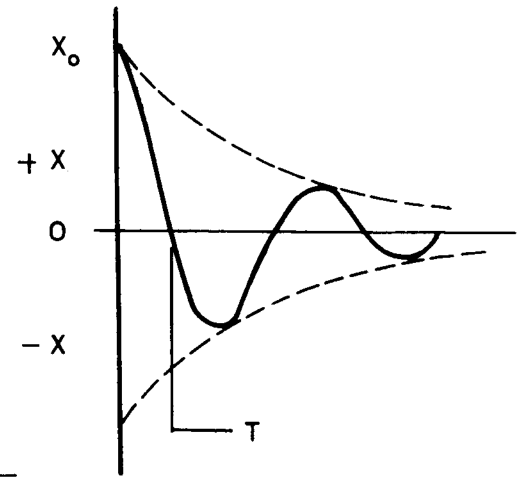
Figure 7. Computer Data on Piston Damped Seal, Radial Clearance = 0.04 inch

TABLE 1

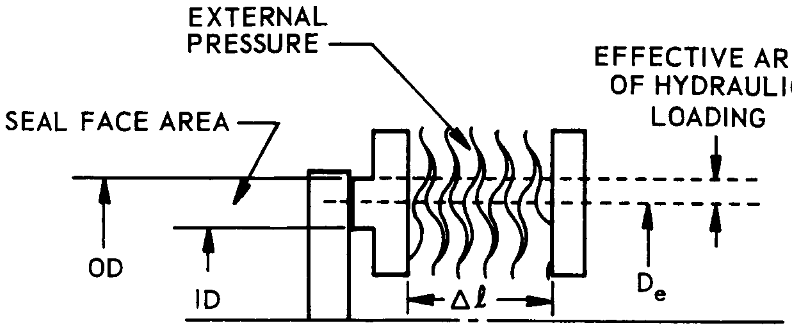
COMPUTER RESULTS, PISTON SEAL

M, lb sec^2/in	K, lb/in	RC, inches	RB, inches	RN, inches	H, inches	Xo, inches	WN cps	C/C_c X10^4	T, Millesiconds
0.00065	400	1.900	1.800	1.840	0.002	0.020	125	1540	4.5
					0.003			450	2.7
					0.004			190	2.3
					0.005			99	2.18
					0.010			12	2.05
					0.020			1.5	2.04
					0.005	0.050		99	2.39
					0.010			12	2.08
					0.020			1.5	2.04
	500				0.005		139	88	2.14
					0.010			11	1.86
					0.020			1.4	1.83





analogy is normally used in considering seal operation; therefore, a change in deformation changes the hydraulic area. The effective hydraulic area is a computed value, normally in terms of the effective diameter (De) as shown below.



Computation of the effective diameter in this case does not include the pressure distribution across the seal face width because during tests to obtain seal face loading data, the face is coated with an adhesive to establish a known sealing point at the seal face OD.

Therefore, the effective hydraulic area is

$$\pi/4 \left(OD^2 - D_e^{\,2}\right) = \frac{Fh}{P}$$

where

Fh = hydraulic force

P = imposed pressure

$D_e = \sqrt{OD^2 - \frac{4Fh}{\pi P}}$

For computation of the seal face diameters a balance ratio is required which is expressed as

$$B = \frac{\text{effective hydraulic area}}{\text{seal face area (Af)}}$$

A balance ratio of 0.7 was chosen for these seal designs to provide a margin for dimensional tolerances and variation of pressure distribution across the seal face.

Based on hydraulic force data obtained from the seal vendor, the seal face diameters are computed relative to the established balance ratio of 0.7. For this case, the diameters chosen are:

$$\text{OD } 3.680 \begin{smallmatrix} +0.000 \\ -0.002 \end{smallmatrix}$$

$$\text{ID } 3.516 \begin{smallmatrix} +0.002 \\ -0.000 \end{smallmatrix}$$

ORIFICE DAMPED SEAL

Figure 8 depicts the final design of the orifice damped seal. Listed below are the control parameters governing operation of the seal.

$M_1 = 0.0009 \dfrac{\text{lb-sec}^2}{\text{in.}}$

$M_2 = 0.0012 \dfrac{\text{lb.-sec}^2}{\text{in.}}$

$A_1 = 4.1$ sq in.

$A_2 = 5.0$ sq in.

$A_3 = 0.20$ sq in. to 0.05 sq in.

$N = 15$ to 25

$K_1 = 500$ lb/in.

$K_2 = 2000$ lb/in.

where

M_1 = effective mass of seal and fitting

M_2 = effective mass of compensator end fitting

A_1 = area of seal cavity

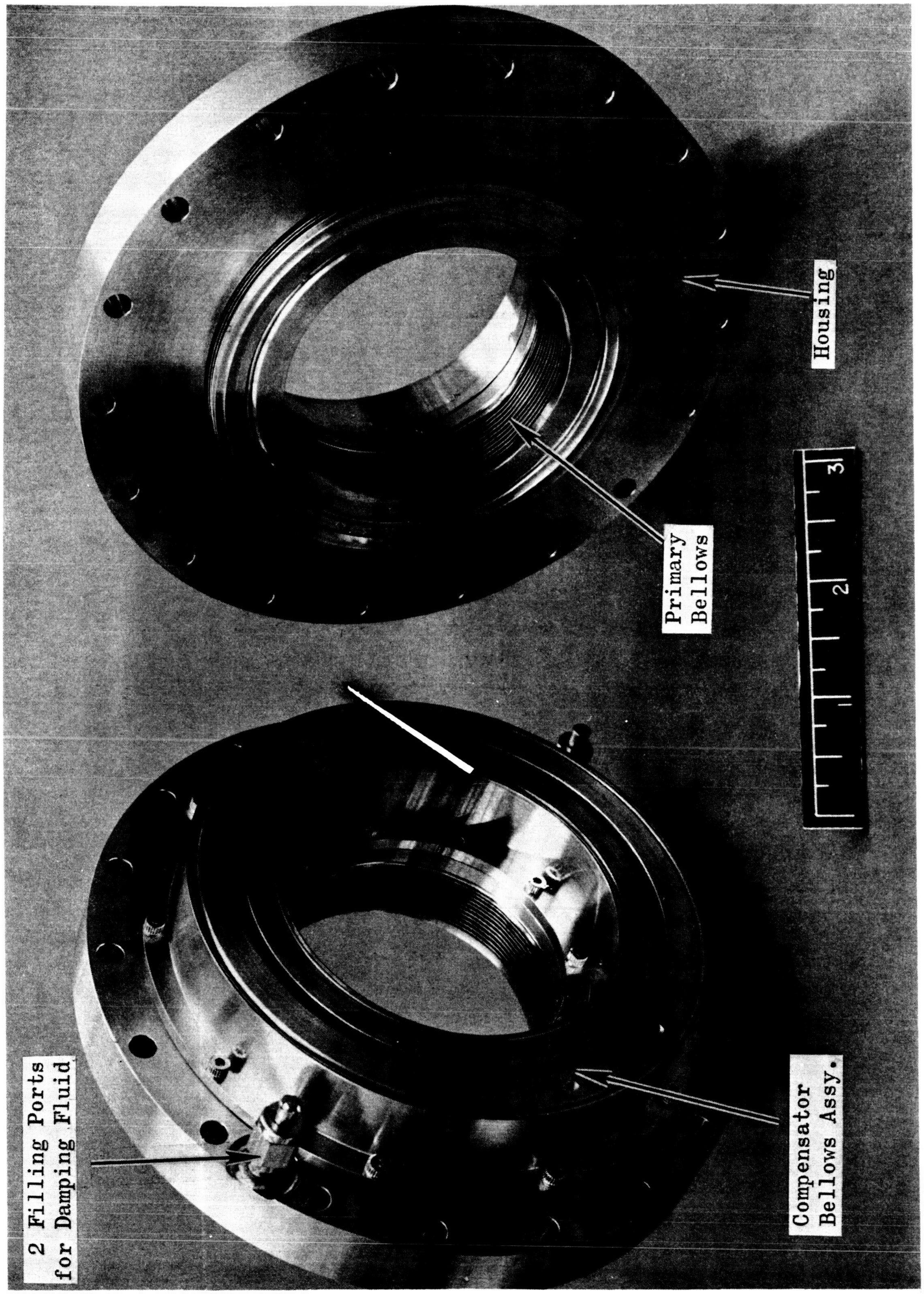


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Figure 8. Orifice Damped Seal

A_2 = area of compensator cavity

A_3 = total area of orifice

N = number of holes

K_1 = spring rate of seal bellows

K_2 = spring rate of compensator bellows

The generalized equations solved for the fluid conditions of laminar and turbulent flow are

$$\frac{d^2x}{dt^2} + Kx + \frac{C\,dx}{dt} = 0 \text{ for laminar}$$

and

$$\frac{d^2x}{dt^2} + Kx + C\left(\frac{dx}{dt}\right)^2 = 0 \text{ for turbulent}$$

Some of the results of the computer study varying the control parameters are shown in Table 2.

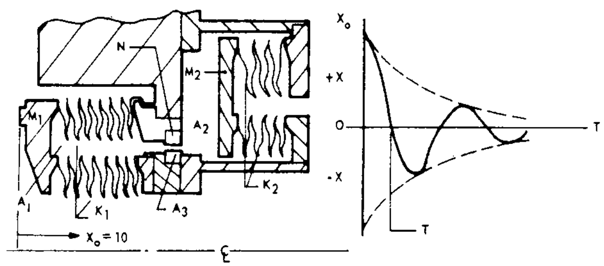
Computer analysis indicates damping can be obtained with the described system using liquid nitrogen; however, a stable condition of liquid is required during all periods of operation. Sodium potassium, used as the damping medium, is a more predictable fluid especially in a temperature environment of 1000 F or greater.

Sodium potassium is a eutectic alloy called NaK - 77 (77-percent K and 23-percent Na) and is being used as the damping medium in the orifice damped seals for operation in a 1000 F gaseous environment. A review of the present day hydraulic fluids revealed that they would not be as satisfactory as NaK over the temperature range of interest, 70 to 1000 F.

TABLE 2

COMPUTER RESULTS, ORIFICE SEAL

M_1 lb sec^2/ sq in	M_2 lb sec^2/ sq in	A_1 sq in	A_2 sq in	K_1 lb/sq in	K_2 lb/sq in	A_3 sq in	N	Do, inch	T(l) milli-second	T(t) milli-second	Wn, cps	C/Cc (t) X10^4	C/Cc (l) X10^4
0.0009	0.0012	4.1	5.0	500	2000	0.20	15	0.132	1.34	1.46	190	1.00	1.68
	NaK DAMPING MEDIUM					0.10		0.092	1.55	2.19	164	3.47	8.22
						0.05		0.066	1.91	5.25	134	11.30	37.87
						0.20	20	0.112	1.34	1.47	190	1.34	1.94
						0.10		0.080	1.55	2.29	164	4.62	9.49
						0.20	25	0.100	1.34	1.49	190	1.67	2.17
						0.10		0.070	1.55	2.39	164	5.78	10.61
						0.05		0.050	1.91	1.59	134	18.84	48.89
						0.20	15	0.132	1.36	1.49	187	0.28	1.82
	LN_2 DAMPING MEDIUM					0.10		0.092	1.59	2.27	160	0.97	8.82
						0.05		0.066	1.97	5.57	129	3.14	40.32
						0.20	20	0.112	1.36	1.51	187	0.38	2.10
						0.10		0.080	1.59	2.39	160	1.29	10.18
						0.20	25	0.100	1.36	1.53	187	0.47	2.35
						0.10		0.070	1.59	2.49	160	1.61	11.39
						0.05		0.050	1.97	1.57	129	5.24	52.06



NOMENCLATURE

(l) = LAMINAR
(t) = TURBULENT
N = NO. OF HOLES
Do = DIAMETER OF ORIFICE

The properties of interest in this program are wide operating temperature range, low density, high thermal conductivity, and relatively good chemical stability. NaK-77 has a melting point of 12 F and boils at about 1443 F under atmospheric pressure. The density is comparable to that of conventional hydraulic fluids, while its viscosity is somewhat lower than that of water. Figures 9 through 13 show the vapor pressure, viscosity, density, thermal conductivity, and specific heat of NaK-77 as a function of temperature.

Because of the hazards imposed when using NaK, safety precautions are required. Listed below is an outline to show the response of NaK to certain environments.

1. Water reacts violently with NaK, to form hydrogen gas and the oxides and hydroxides of sodium and potassium. The heat of reaction is great enough to ignite mixtures of hydrogen and oxygen if air is present.
2. Alcohols react mildly with NaK at room temperature and may be used for cleaning under controlled conditions. Reaction is fastest with methyl and ethyl alcohol, slower with the heavier alcohols such as propyl and butyl. Air must be excluded from the cleaning system.
3. At room temperature, bulk NaK open to the air reacts slowly with oxygen to form a surface scum. If spilled, small particles of NaK may ignite spontaneously, particularly with dust and many combustible materials. The ignition temperature for bulk NaK in air is about 400 F.
4. Carbon tetrachloride reacts violently and explosively with NaK.

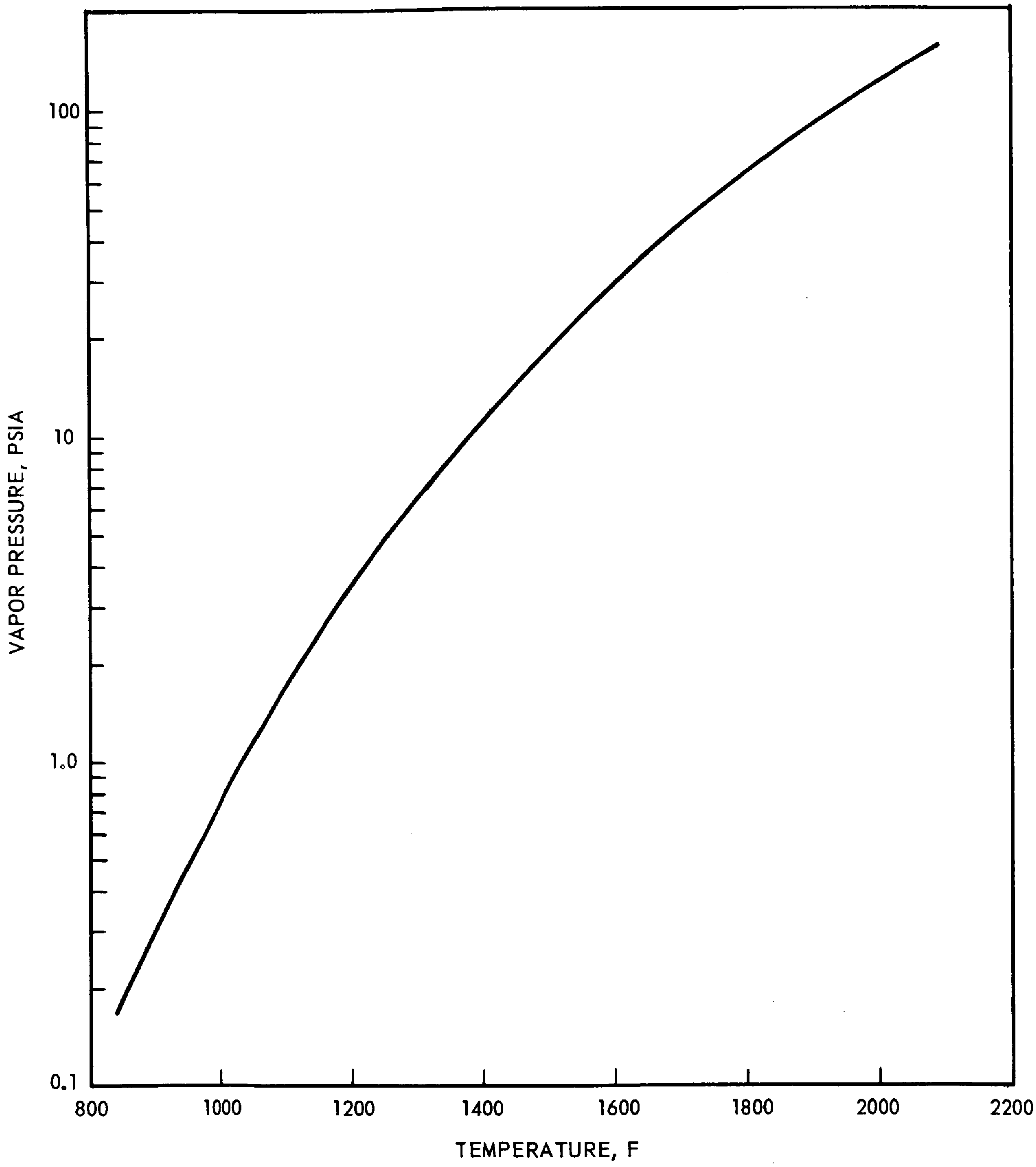


Figure 9. Vapor Pressure vs Temperature, NaK-77

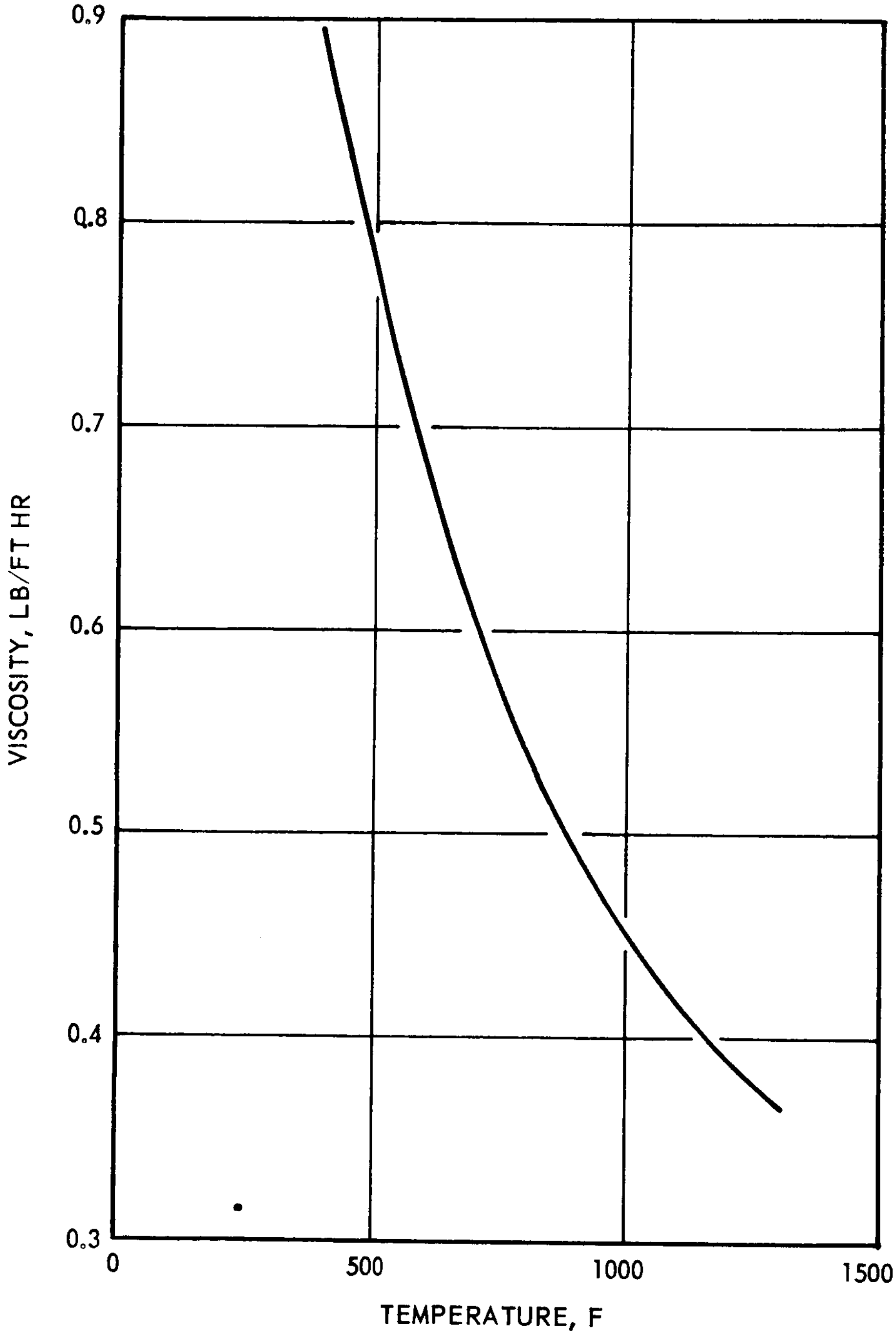


Figure 10. Viscosity vs Temperature, NaK-77

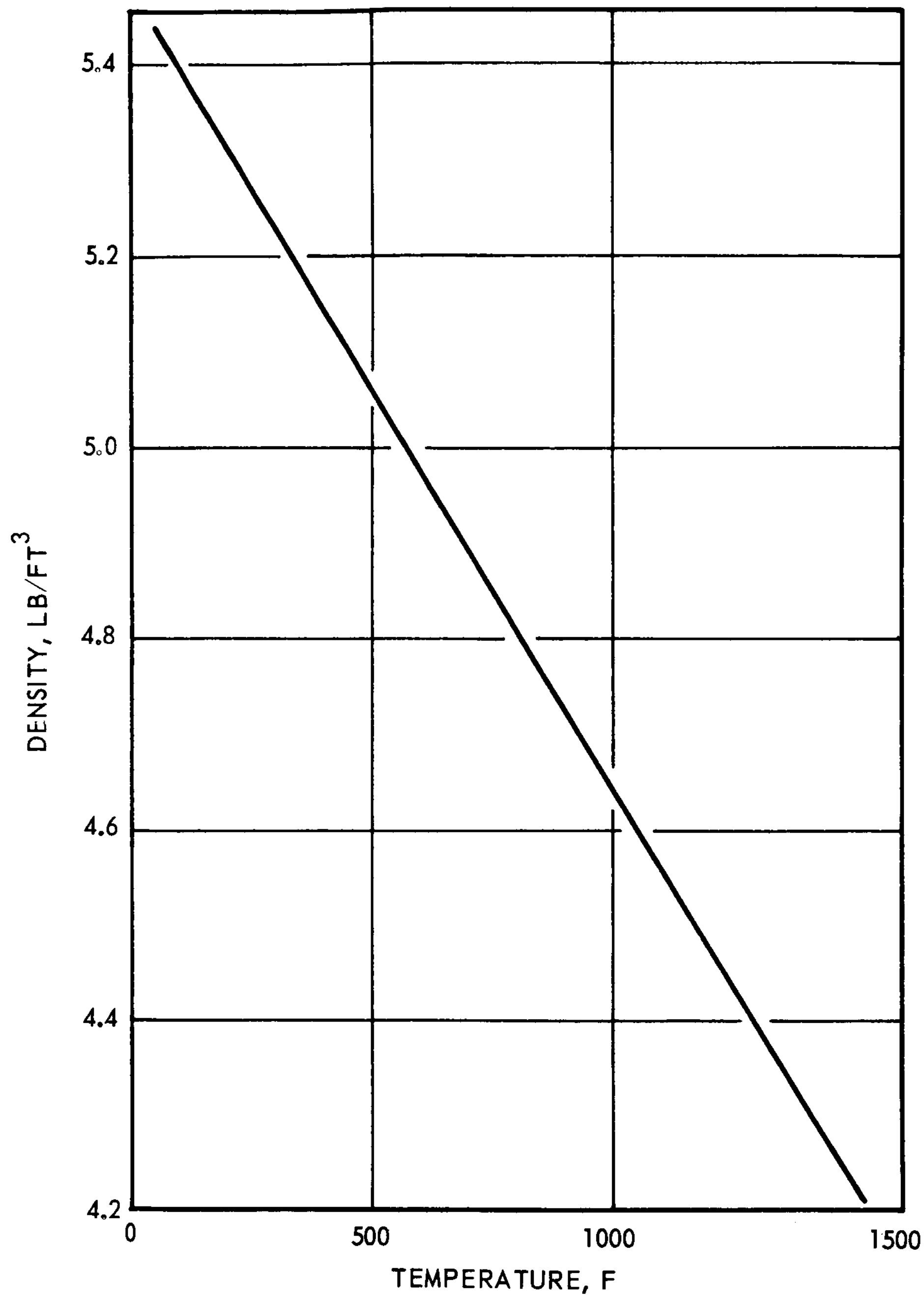


Figure 11. Density vs Temperature, NaK-77

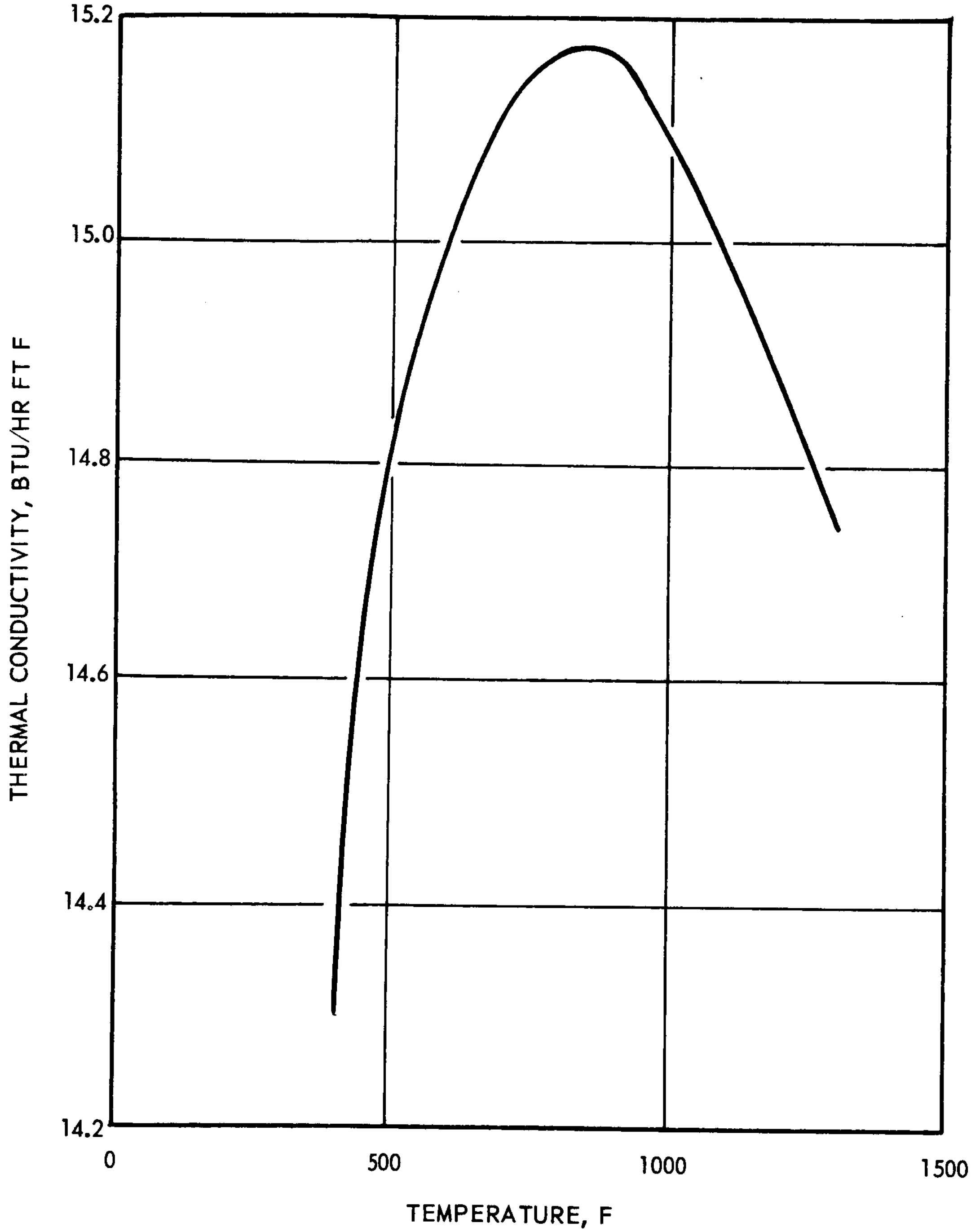


Figure 12. Thermal Conductivity vs Temperature, NaK-77

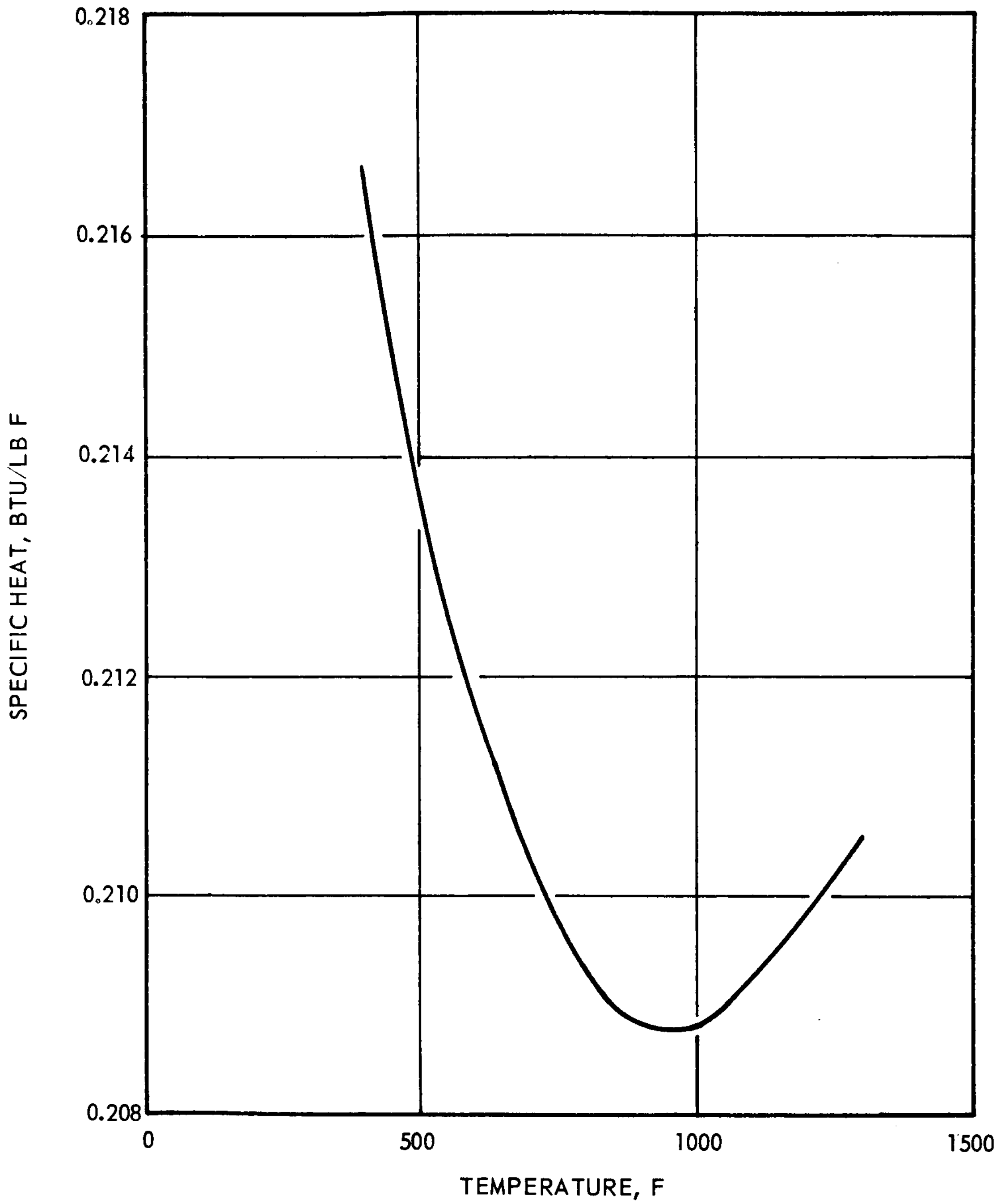


Figure 13. Specific Heat vs Temperature, NaK-77

5. Trichloroethylene froms highly unstable, dangerously explosive dichloroacetylene.

6. Natural hydrocarbons from refined petroleum oil are inert. Kerosene and white mineral oil are very useful in flusing the system and excluding air from the surface of NaK in open containers.

7. Carbon dioxide is considered dangerously reactive.

8. Fire extinguishing materials are dry calcium carbonate, dry sand, and dry sodium chloride.

Corrosive attack of NaK on Inconel 750 was considered to be a problem in high-temperature testing, one requiring investigation. There are several important modes of attack.

One is mass transfer, in which elements from one metal in one part of the system are dissolved and reprecipitated in another part of the system. Grain boundaries are also attacked through fingerlike dissolution along the boundaries, uneven surfaces, or as slivers of corrosion product in between grains. Dissolution of stringers of inclusions is a mode of attack of particular importance with welded bellows plates. Plate stock rolled in a "dirty" condition or else welded without proper cleaning will have soluible inclusions in the bellows plate as well as in the weld bead. NaK will dissolve these stringers and weaken the structural joint. Stress corrosion is also a potential failure mode with the welded bellows seal.

An Inconel 750 TIG welded bellows specimen, made under current state-of-the-art of seal vendor weld specifications, was submerged in NaK for 7 hours at 1000 F. The bellows specimen was compressed to induce tensile stresses in the bellows. Figures 14 and 15 show the test specimen after

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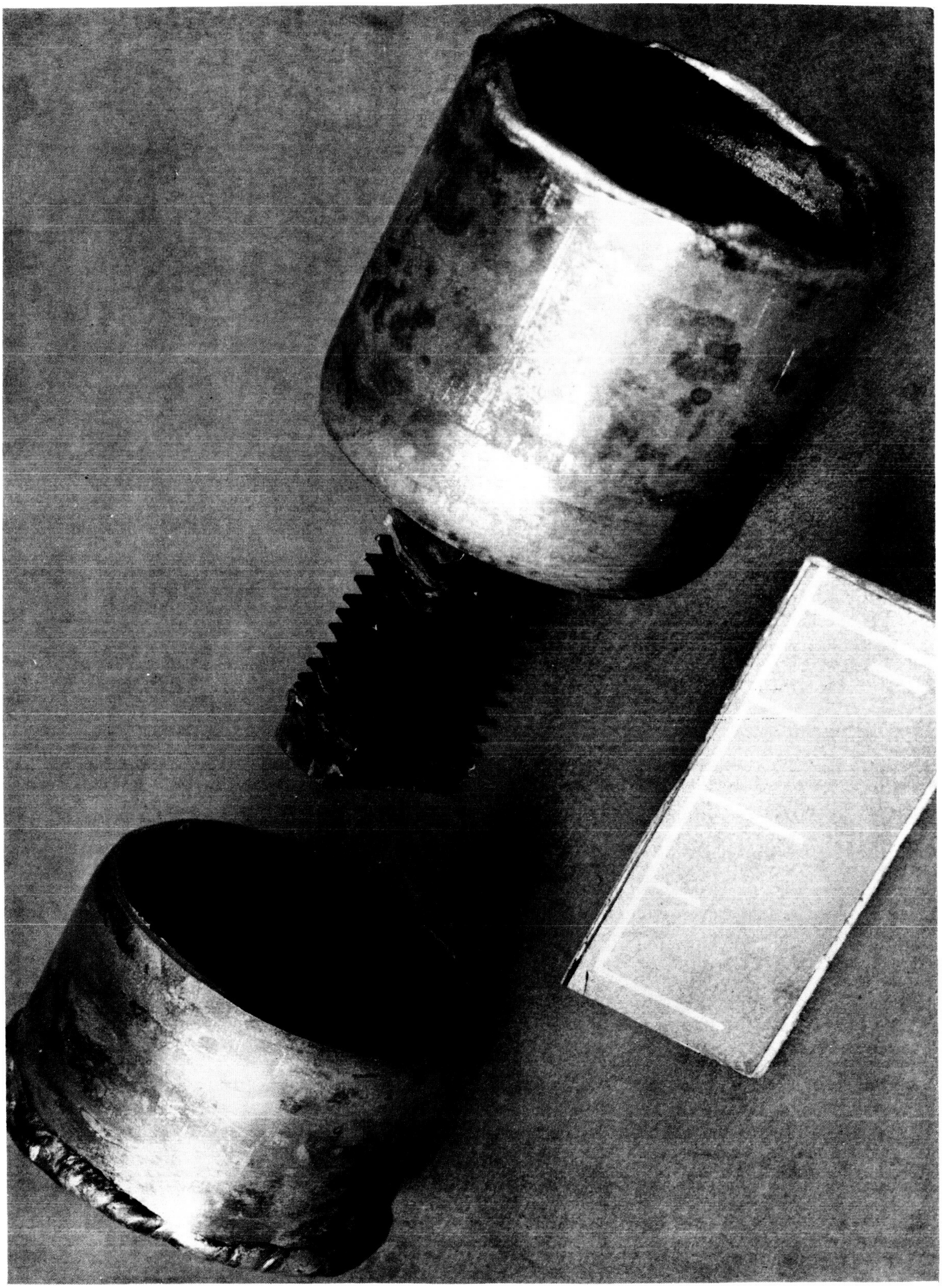

Figure 14. NaK Exposed Bellows (View A)

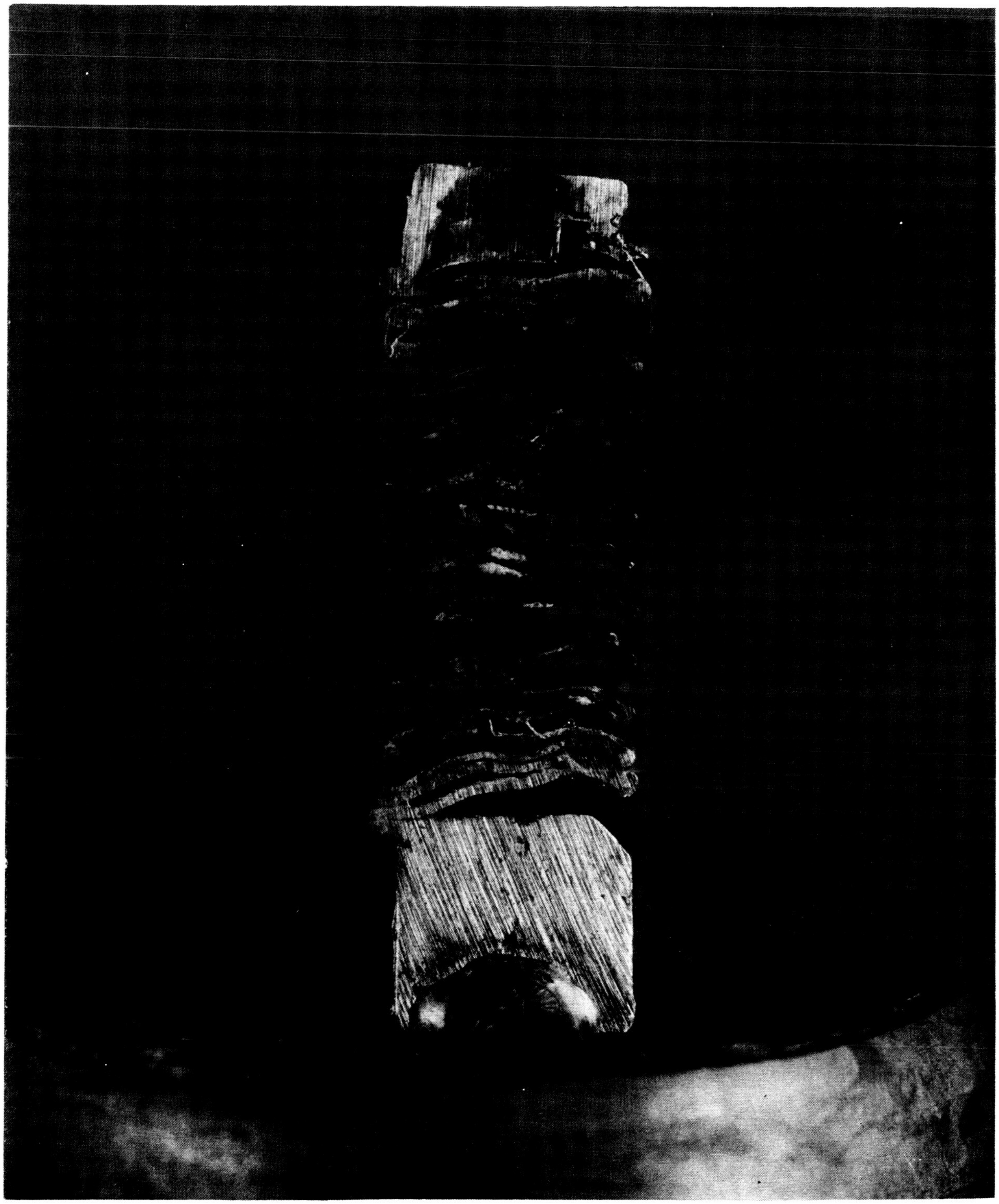

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Figure 15. NaK Exposed Bellows (View B)

exposure. The bellows is Inconel 750, the end fittings are Inconel 600, and the can enclosure is Inconel 750. The can enclosure with the specimen and NaK was welded shut in an argon atmosphere. The residue between the bellows plates as shown in Fig. 15 is oxide scum formed when the specimen was cleaned with water after exposure.

Posttest inspection revealed the bellows to be satisfactory. Mass transfer and stress corrosion cracking were not observed; however, some dissolution did take place along the grain boundaries of the Inconel 600 end fitting. Oxide scale formed during welding in the bead root was dissolved. This attack was confined to the outer surface but does stress the importance of "clean" welds.

PARTICLE DAMPED SEAL

Initially, a feasibility study was conducted to determine the damping potential of metal powder. Molybolenum was selected on the basis of availability.

Various sizes of spherical powder was purchased and a simple test model was fabricated. The test model consists of two aluminum blocks with a series of drilled holes to contain the spherical particles. The blocks are clamped together on a spring beam and mounted to the vibration plate as shown in Fig. 16.

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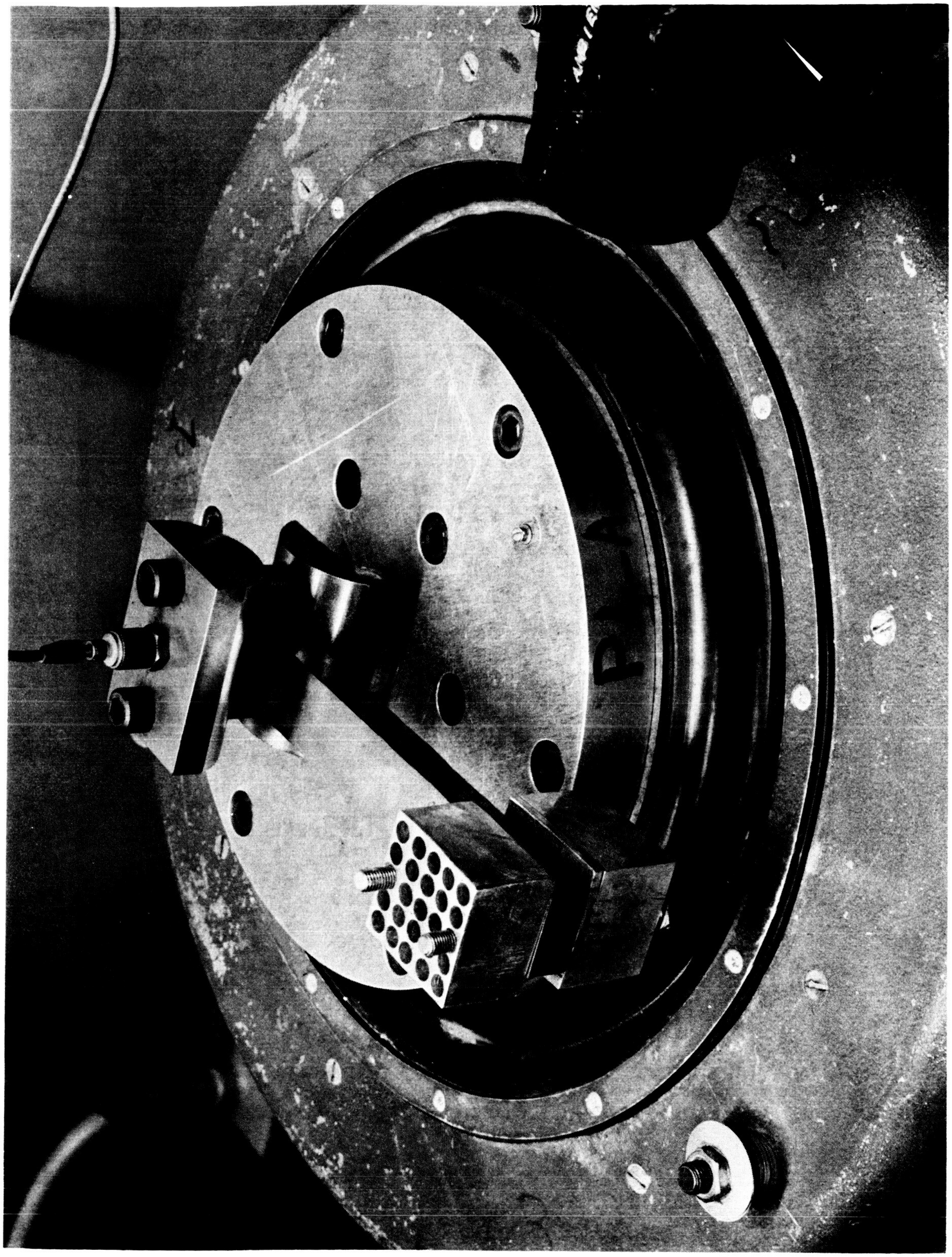


Figure 16. Particle Damping Test Setup

Table 3 gives the characteristics of the particles tested:

TABLE 3

SIZE, WEIGHT, AND DENSITY OF
SPHERICAL MOLY PARTICLES
(SIX SAMPLES CONSIDERED)

Sample No.	Diameter		Weight, Grams	Approximate Density, lb/in.^{3}
	Inches	Microns		
1	+0.0059	149	276.75	0.232
2	0.0059/0.0043	149/131	184.50	0.230
3	0.0043/0.0035	131/88	982.65	0.225
4	0.0035/0.0029	88/74	149.00	0.225
5	0.0029/0.0023	74/57.5	251.15	0.226
6	0.0023/0.0017	57.5/44	245.90	0.226

NOTE: The quantity of particles above represents the quantity available for the test and not the amount actually used.

Two methods of determining the amount of damping were employed: (1) forced vibration, and (2) free vibration.

Forced Vibration

The exciting force is held constant (2 g-peak) and the exciting frequency is varied. The damping is determined by measuring the bandwidth of resonance peak at the half-power points in terms of normalized frequency. The formula is as follows:

$$\Delta f \equiv \text{damping} \equiv 3.14 \left(\frac{f_2}{f_n} - \frac{f_1}{f_n} \right)$$

where

f_n = resonant frequency

f_1 = lower half-power frequency

f_2 = upper half-power frequency

Free Vibration

One end of the beam is fixed and the other end is deflected and then allowed to freely return to its starting reference point. The damping is determined from the rate of decay of oscillation. This method is defined as the logarithmic decrement, and it is the natural logarithm of the ratio of any two successive amplitudes:

$$\Delta \equiv \text{damping} \equiv \ln \frac{X_2}{X_1}$$

where

ln = natural logarithm

X_1 = first cycle amplitude

X_2 = second cycle amplitude

The forced vibration method of testing was the most successful and reliable. Because of the loss of energy to the beam support, the data from the free vibration method was inconsistent.

In addition to damping, the frequency at resonance and the amplification at resonance were measured and recorded. The amplification is defined as the ratio of the acceleration into the beam to the acceleration out of the beam.

$$\text{Amplification} = \frac{\text{G out}}{\text{G in}} \quad \text{or} \quad \frac{\text{Ao}}{\text{Ai}}$$

Tests were run on all six particle sizes with the fixture half-full and on four particle sizes with the fixture full. There were not enough particles of the No. 2 and No. 4 sizes to fill the fixture. The results of the tests are as follows:

Particle	Half Full				Full			
	Fn	Ao/Ai	Δ f	Δ	Fn	Ao/Ai	Δ f	Δ
No. 1	165	10.80	0.368	0.288	155	4.51	0.735	0.523
No. 2	164	11.00	0.282	0.157				
No. 3	163	8.91	0.325	0.214	143	3.38	1.156	0.521
No. 4	162	7.36	0.416	0.273				
No. 5	162	6.49	0.543	0.300	149	3.61	1.081	0.826
No. 6	159	5.82	0.663	0.434	149	3.87	0.950	0.877
Empty	170	37.80	0.100	0.070				

Fn = resonant frequency

Ao/Ai = amplification

Δ f = forced vibration damping

Δ = free vibration damping

The results of the test clearly indicate that the best damping characteristics are obtained with the No. 6 size particles. As the particle diameters decreased, from size No. 1 to Size No. 6, the resonant frequency decreased, the amplification decreased, and the damping from both the free and the forced vibration increased.

A test was conducted to determine what portion of the damping was caused by the mass of the particles and what portion was caused by the motion of the particles.

Two steel masses (equal to the particle mass) were attached to the empty particle holders and the test conducted in the same manner as the previous tests. The test was repeated with one of the steel masses removed. The results shown below indicate that the masses caused the resonant frequency to decrease, the amplification to increase, and the damping to decrease.

Weight, grams	Fn, cps	Ao/Ai	Δ F	Δ
91.5	144	59.8	0.054	0.094
185.5	129	50.0	0.074	0.094

Comparing the test results further indicates that damping can be attributed to relative movement of the particles in the container and that a solid mass of particles would have no effective damping.

Damping was next measured with the No. 6 particle size varying the quantity by weight from 50 to 200 grams. This was done to determine the effects of different amounts of the same particle size on damping. Figures 17 and 18 show the results of the tests from forced vibration and free vibration.

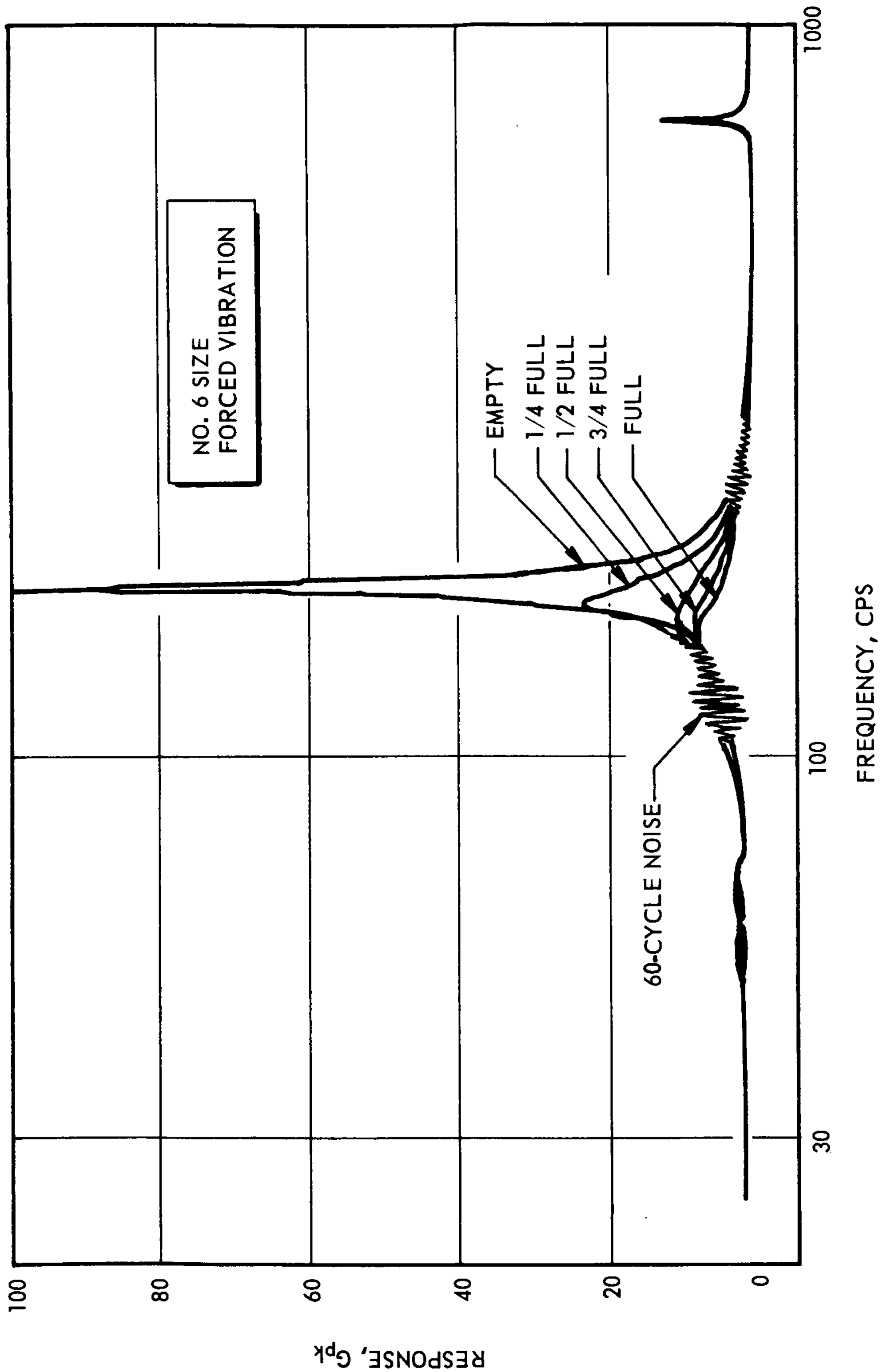


Figure 17. Particle Vibration Test

TEST PROGRAM AND HARDWARE DESCRIPTION

TEST PROGRAM

The seals in this program were not designed for a specific turbopump application or planned for rotational tests. However, requirements and operational criteria are based on conditions consistent with advanced turbomachinery. The following tests were specified.

Static Leakage and Proof Pressure

Static leakage and proof pressure tests are conducted at GN_2 pressures up to 375 psi to verify mechanical integrity and design requirements. Leakage across the seal face as well as a mass spectrometer check (where applicable) for porous weld beads is made.

Total Face Loading

Total load tests are conducted to measure the effective seal face unit loading as a result of combined bellows spring load and either pneumatic or hydraulic loading. This is conducted using an Instron machine having a rate of stroke range from 0.0003 ips to 0.83 ips which will allow a dynamic measure of seal face loads.

Pressure Cycling

Pressure cycling is a measure of the bellows integrity to withstand pressure pulses up to 200 ±50 psig in a liquid nitrogen environment. Seal leakage is measured throughout the test. Tests at 1000 F are not conducted because of difficulties in designing a pressure pulsing system at this temperature. Over 10^6 cycles were planned.

Mechanical Cycling

Mechanical cycling tests are conducted to monitor seal integrity when exposed to displacements of ±0.030 and ±0.015 inches at 16 to 100 cps in LN_2 and 1000 F GN_2. Seal leakage and displacement rate was recorded for a period of 10^6 cycles.

Recovery Rate

The recovery rate test involves displacement of the mating ring from the maximum design bellows compression of the seal to the minimum compression with the intent to show the ability of the seal face to follow the mating ring. Seal leakage is also observed.

Vibration

Vibration tests consist of the following:

1. Resonance Search. Record the frequency of all resonant points observed in a 2 g peak-to-peak sweep from 15 cps to 2000 cps in the axial axis of the seal.

2. Sinusoidal Resonance Test. Subject the seal to its major resonant frequency for 10 minutes in the axial axis at the following levels:

   15 to 50 cps, at 0.2-inch double amplitude
   50 to 500 cps, at 12 g peak
   500 to 1000 cps, at 0.0006-inch double amplitude
   1000 to 2000 cps, at 30 g peak

   NOTE: $g = 0.0511\ f^2d$ (gravity units)

   where

   f = frequency in cps
   d = double amplitude in inches

HARDWARE DESCRIPTION

Mechanical Cycle Tester

The mechanical cycle and recovery rate tester Fig. 20 has the capability of mechanically cycling the seals of this program at ±0.015 to ±0.030 inch amplitude at 16 to 100 cps. The test housing contains eight 525-watt cartridge heaters to maintain a 1000 F gas environment at the seal OD while pressurized up to 250 psig. The test housing also contains ports for filling and maintaining LN_2 in the cavity exposed to the seal. Thermal insulation surrounds the test housing to reduce heat losses.

The piston housing has two cavities separated by a piston; the end cavity can be pressure pulsed to move the shaft for recovery rate studies.

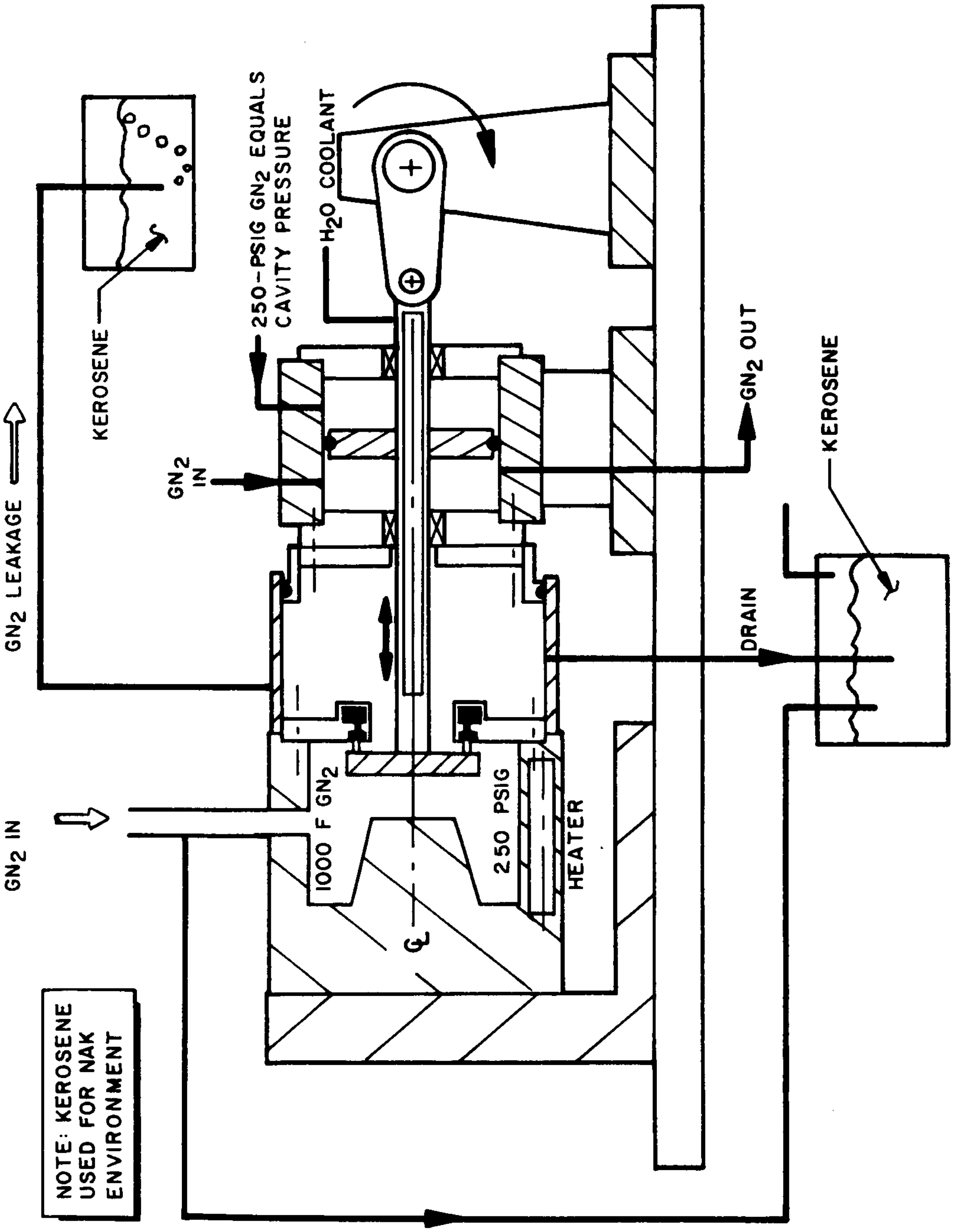


Figure 20. Mechanical Cycling Tester

Vibration Tester

The vibration fixture consists of a housing and mounting face for the seal. The test housing contains heaters and LN_2 ports to be used in the mechanical cycling test. The back side of the seal can be fitted with a cap to measure leakage or without the cap for visual observation of bellows vibration and seal response. Proof pressure and static leakage tests can also be made in this fixture.

Total Load Tester

The total load tester consists of a mounting fixture for the seal adapted to be mounted in an Instron test machine to measure the total seal face load within operating pressure and the displacement.

Steady-state pressures are measured with standard gages; pressure pulses are measured with pressure transducers. The pressures are those primarily imposed on the seal during test. Thermocouples are used to monitor the temperature close to the seal face. Seal leakage, either LN_2 or 1000 F GN_2, is ported through a heat exchanger to a flowmeter. Accelerometers are used to measure displacement input and output in addition to linear transducers where applicable.

Table 4 specifies the instrumentation used for the tests.

TABLE 4

INSTRUMENTATION

Test	Measurement	Location	Operating Range	Measurement Device
Mechanical Cycling	Seal Displacement	Bellows ID	±0.015 and ±0.030 inch	Linear Displacement Transducer and Recorder
	Seal Temperature	Bellows OD Cavity	-32 F and + 1000 F	Thermocouple
	Cycling Rate	Seal Face and Tester Shaft	16 to 100 cps	Cycle Recorder
	Seal Pressure GN_2 and LN_2	Bellows OD Cavity	200 psig to 250 psig	Direct Reading Gauge
	Seal Leakage	Seal Cavity	0 to 1000 scims	Flow Meter
Recovery Rate	Shaft Motion	Shaft	0.090 to 0.150 inch seal height	Linear Displacement Transducer and Recorder
	Seal Temperature	Bellows OD Cavity	-323 F and +1000 F	Thermocouple
	Seal Pressure	Bellows OD Cavity	200 to 250 psig	Direct Reading Gauge
Pressure Cycling	Seal Pressure LN_2	Bellows OD Cavity	200 ±50 psig	Pressure Transducer and Recorder
	Seal Temperature	Bellows OD Cavity	-323 F	Thermocouple
	Cyclic Displacement	Seal Face OD	Recorded Amplitude	Cycle Counter and Amplitude Readout
	Seal Leakage	Seal Cavity	0 to 1000 scims	Flow Meter

TABLE 4

(Concluded)

Test	Measurement	Location	Operating Range	Measurement Device
Total Load	Total Load	Seal Face	0 to 500 pounds	Load Cell
	Seal Height	Seal Face	0.150 to 0.090 inch	Dial Indicator
	Seal Pressure GN_2 and LN_2	Bellows OD Cavity	0 to 250 psig	Direct Reading Gauge
	Rate of Seal Compression	Seal Face	0.050 to 0.830 ips	Instron Console
	Seal Leakage	Seal Cavity	0 to 1000 scims	Flowmeter
Vibration	Seal Pressure	Bellows OD Cavity	200 to 250 psig	Direct Reading Gauge
	Vibration Input	Test Housing	0 to 30 g	Accelerometer
	Seal Displacement	Seal Face	Recorded output	Accelerometer/Linear Transducer
	Seal Temperature	Bellows OD Cavity	-323 F to 1000 F	Thermocouple
	Seal Leakage	Seal Cavity	0 to 1000 scfim	Flowmeter

TEST PROCEDURE AND RESULTS

MECHANICAL CYCLING TESTS

Upon completion of dimensional verification, static leakage, and pressure tests, the piston damped and orifice damped seals were mechanically cycled. The particle damped seals were not exposed to this test because sufficient data was obtained from the piston damped seal both having the same bellows configuration.

Piston damped S/N 3 was installed in the tester with a damping piston to maintain a radial clearance of 0.004 inch. The bellows was cycled for a total of 10^6 cycles at 16 cps with ±0.025 inch peak-to-peak amplitude in LN_2 at 250 psig.

The seal face satisfactorily followed the cycling shaft at 16 cps as evidenced from the low and constant LN_2 leakage rates and comparisons of displacement transducer traces. Surges in the leakage rates indicate a lag in the response of the seal face with respect to the shaft. The leakage rates during the mechanical cycling test varied between 0.4 and 0.6 scfm.

Two linear displacement transducers were used: one to monitor the cyclic input axial displacements to the tester shaft, and one to monitor the seal face axial movements in response to the inputs. Outputs from the two transducers were recorded on a CEC Recording Oscillograph. A typical output is shown in Fig. 21 which shows that the seal face carrier followed the shaft displacements.

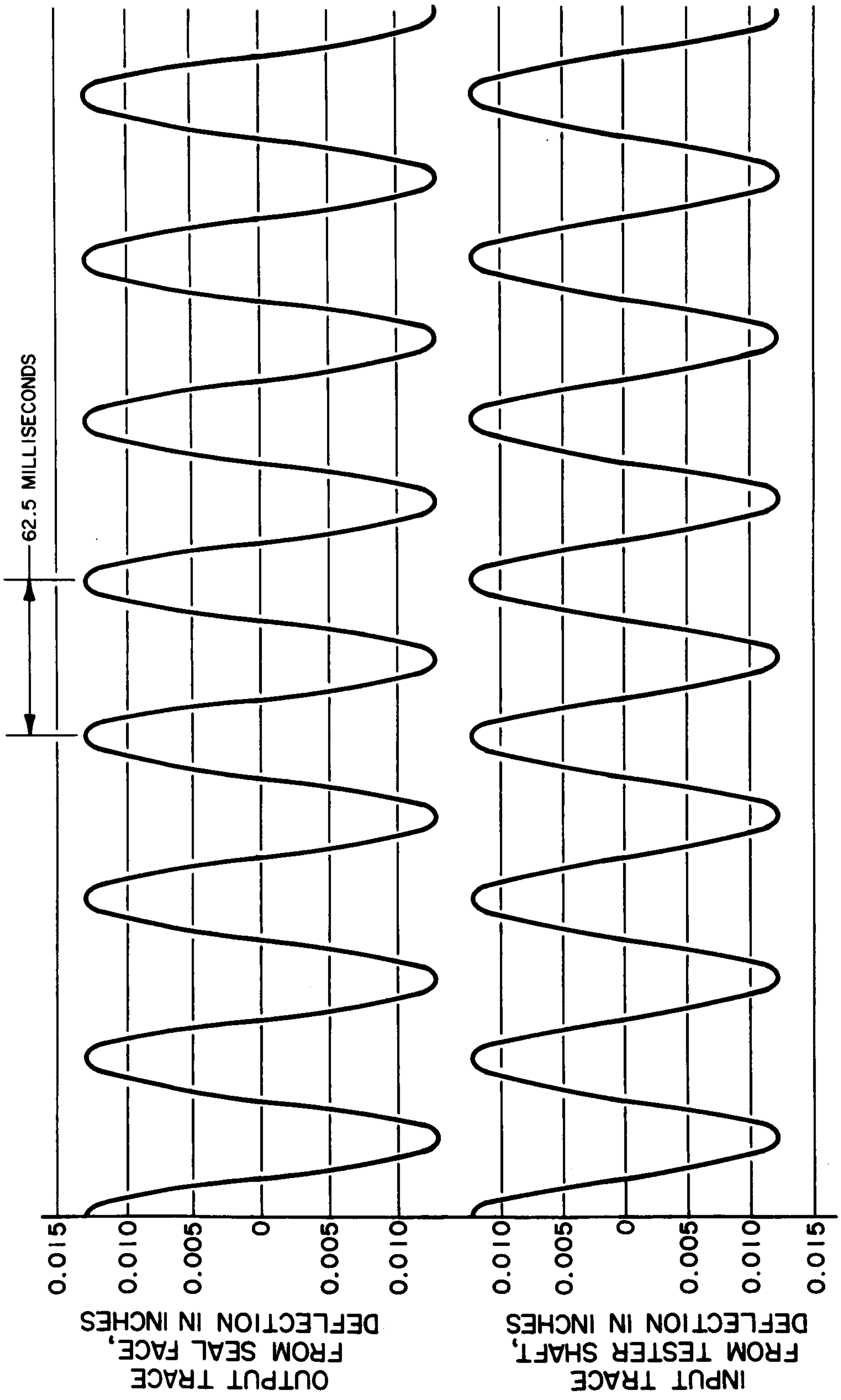


Figure 21. Typical Oscillograph Traces of Transducer Output

Posttest inspection of the seal revealed no failures in the welded bellows. However, welding of the displacement transducer probe tab to the seal face carrier ID had sufficiently distorted the carrier OD to allow rubbing contact between the carrier OD and the piston ring ID. A spring rate test indicated the drag to be only 1 pound.

The next piston damped seal, S/N 2, was mechanical cycled without a piston ring installed to obtain a comparison and to study seal performance without a piston installed.

The piston damped seal, S/N 2, was exposed to mechanical cycles at ±0.007 inch of amplitude at 100 cps in LN_2 at 250 psig. After 69,000 cycles, the seal leakage exceeded the range of the flowmeter and the seal was removed from the tester. Inspection revealed the seal to have a failed bellows in the ID weld at the heat affected zone of the third and eighth convolutions. The seal face apparently did not remain in contact with the mating ring at the 100-cps cycling rate, as evidenced by fretting between the seal face and mating ring and comparison of the displacement transducer traces. Normally, an increase in leakage would be indicative of seal face separation; however, because of the bellows failure, no increase in leakage could be observed.

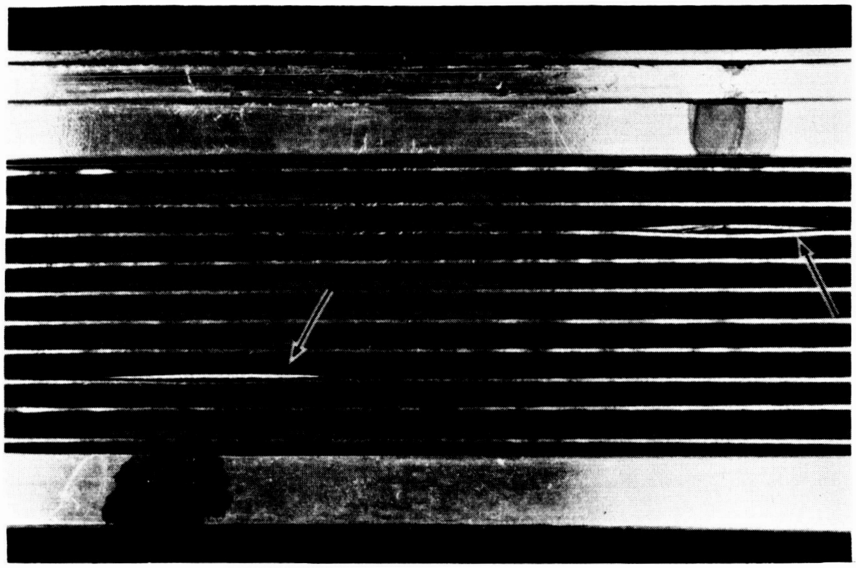
The cycling displacement amplitude was found to be attenuated during the above 100-cps test to ±0.007 inch; the previous displacement amplitude at 16 cps was approximately ±0.013 inch. A spherical bearing was replaced in the linkage system in an attempt to reduce total system play. A checkout test further resulted in failure of the test hardware to the point of bending the tester support and displacement shaft. Because of the apparent limited capability of the tester, the frequency was reduced to 16 cps in all cases.

A metallographic analysis to determine the mode of failure was conducted; the results showed the following:

1. The Inconel X-750 bellows failed as a result of fatigue along the base of the ID convolution welds.
2. The welds of the bellows were metallurgically sound and of acceptable geometry.
3. The parent metal microstructure of the bellows was metallurgically sound and did not contribute to the failure.

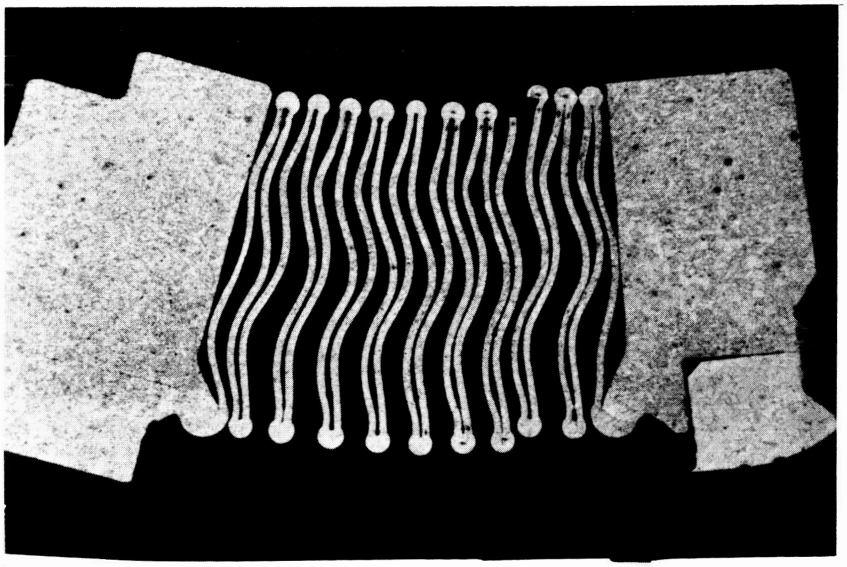
A transverse section of the bellows was removed and photographed to show the relative location of the two failed areas shown in Fig. 22. Sections through each of the two failed areas were prepared for metallographic examination and measurement. A fractograph was taken (Fig. 24) which establishes the failure to be caused by fatigue. Figures 23, 25, and 26 show the extent, nature, and location of the failures.

The bellows plate thickness was measured to be 0.006 inch, and microhardness readings established that the bellows was in the heat-treated condition. The parent metal Rockwell hardness was 35 Rc, and the welds were approximately 37 Rc. All associated factors were favorable for optimum service from the Inconel X-750 bellows.



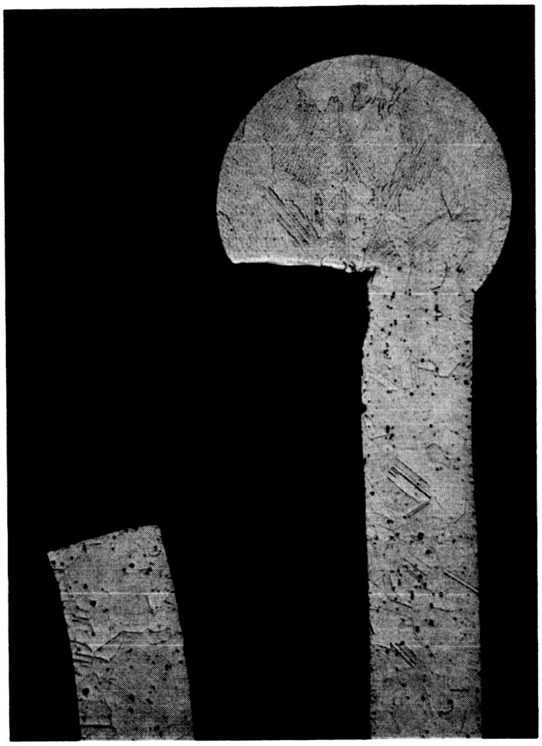
Mag: 3X

Figure 22. Section Removed From Inconel X-750 Bellows Piston Seal Showing Ruptures (arrows) on the ID Surface.



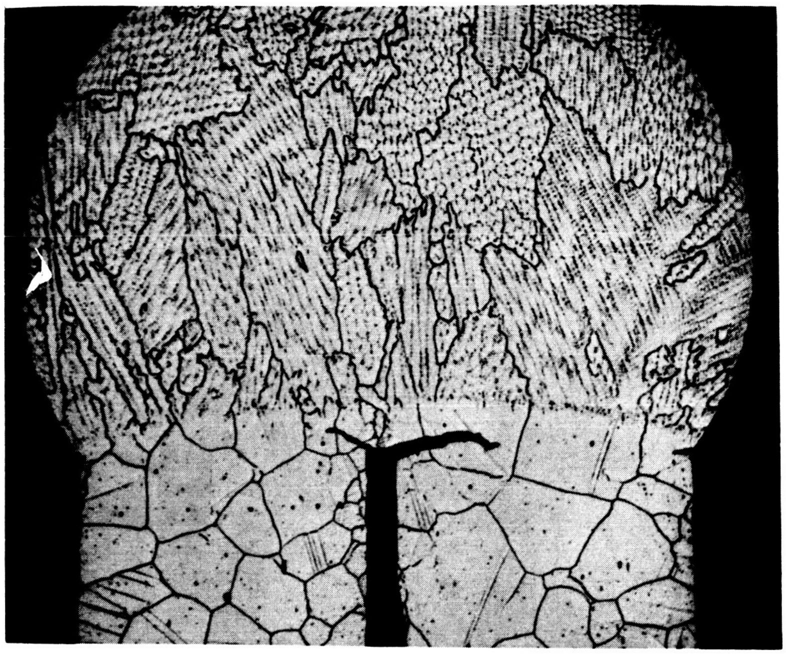
Mag: 8X Etchant: Three Acids

Figure 23. Transverse Section Through Failed Area (Third Convolution From Bottom, Fig. 1).



Mag: 100X Etchant: Three Acids

Figure 24. Fracture Location Along Base of Weld. (Note Straightness of Fracture Which is Typical of a Transgranular Failure)



Mag: 250X Etchant: 10-Percent Oxalic, Electrolytic

Figure 25. Crack Initiation Sites at Natural Stress Risers. (Note clear evidence of transgranular mode of failure)

Mag: 1000X Unetched

Figure 26. Fracture Surface of Failed Inconel X-750 Bellows Showing Conchoidal Markings Which Denote the Progression of the Failure by Fatigue.

In accordance with the program plan, mechanical cycling of the piston damped seals was completed with the inclusion of S/N 4. A tabulated summary of the testing is shown in Table 5.

TABLE 5

TEST SUMMARY, PISTON DAMPED SEALS

S/N	Cycling Rate, cps	Displacement, inch	Piston Clearance, inch	Seal OD Environment	Cycles	Leakage, scfm
3	16	±0.013	0.0042	LN_2 at 250 psig	1,000,000	0.4 to 0.8
3	16	±0.030	0.0042	LN_2 at 250 psig	1,000,000	0.8
2	100	±0.015	no piston	LN_2 at 250 psig	69,000	failure
4	30	±0.030	0.0072	LN_2 at 250 psig	1,000,000	1 to 5

The last test (S/N 4) completed on 22 April showed no change in bellows response when compared with data from previous tests. The higher leakage rate experienced on the last test is not necessarily a function of relative movement between the adjacent sealing surfaces because during all tests, a slight roughening of the contacting surfaces has been noted. Mechanical cycling of orifice damped seals S/N 4 and S/N 2 was completed in March 1966 with no problems at the test conditions shown in Table 6.

The failure of S/N 2 without a piston installed indicates the potential need of a piston, because no failure occurred with S/N 3 or 4. Although the cycling rate was greater for S/N 2, the displacement was only half of that imposed on S/N 3 and 4.

TABLE 6

TEST SUMMARY, ORIFICE DAMPED SEALS

S/N	Cycling Rate, cps	Displacement, inch	Seal OD Environment	Cycles	Leakage, scfm
4	30	±0.030	900 F GN_2 at 250 psig	1,000,000	0.3
2	16	±0.030	LN_2 at 250 psig	1,000,000	< 0.3

Posttest inspection showed the seals to be in excellent condition with only very minor scratches on the seal faces. A static leak check with GN_2 at 30 psig indicated a leakage of less than 10 scims. A typical test setup is shown in Fig. 27 , 28 , and 29.

On 12 May 1966, a mechanical cycling test was completed on seal S/N 6 with sodium potassium (NaK) in the bellows interior. The seal was filled with NaK under an argon blanket. When filled, the bellows were compressed from the free height (0.163 inch) to the design installation height displacing an excess volume of NaK. With the filling ports still open, the bellows were further compressed to displace an additional volume of 7 cc to account for a 11.5 percent volume increase of NaK when heated from 60 to 900 F. The ports were closed at this point leaving a volume of 52.5 cc remaining from an original volume of 69.5 cc.

The seal was installed in the tester at a compressed height of 0.120 inch. To verify the integrity of the closed system, the tester heater rods were energized and the seal heated to 900 F. The drive motor was started which

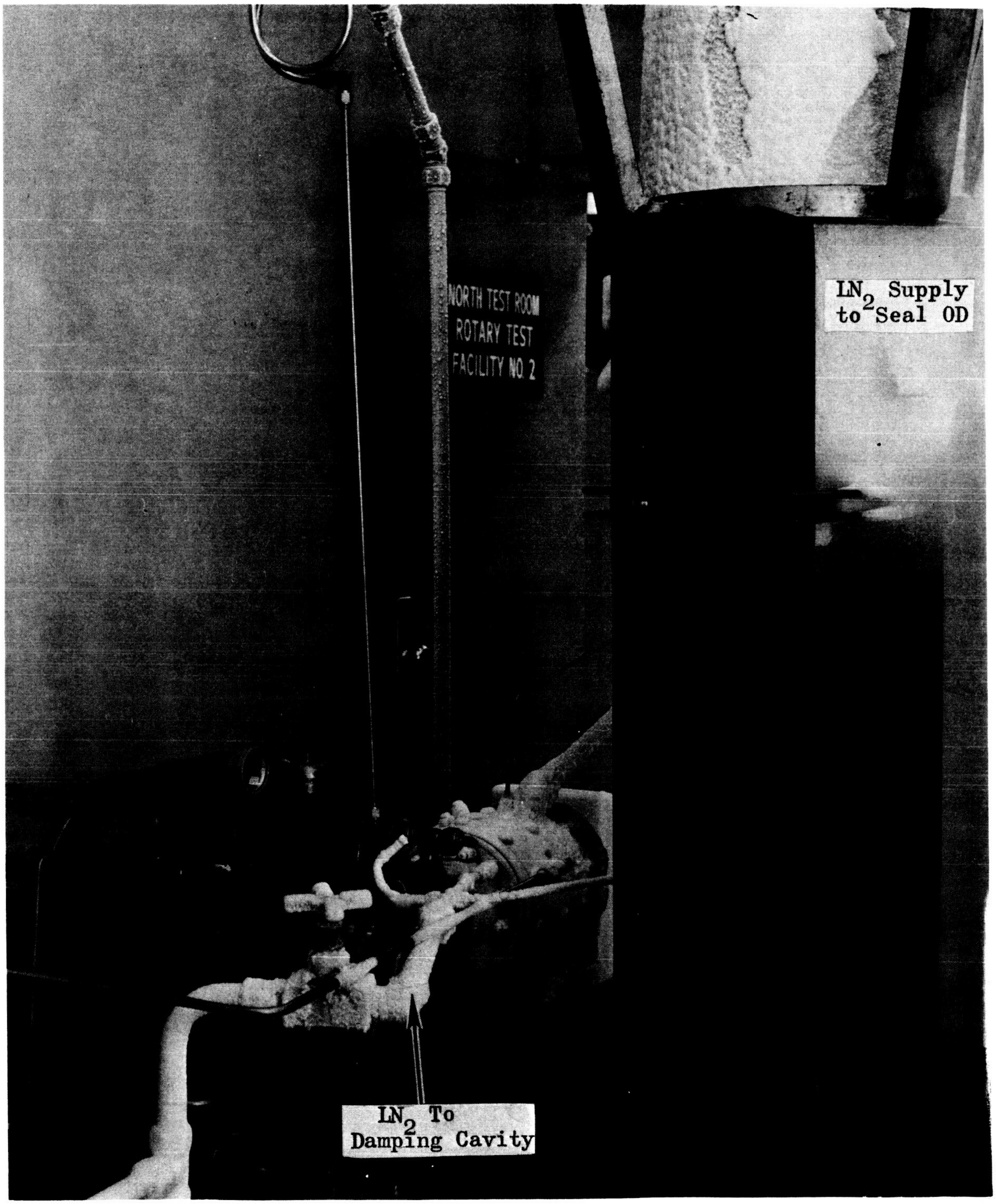


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Figure 27. Test Setup, Orifice Damped Seal (View 1)

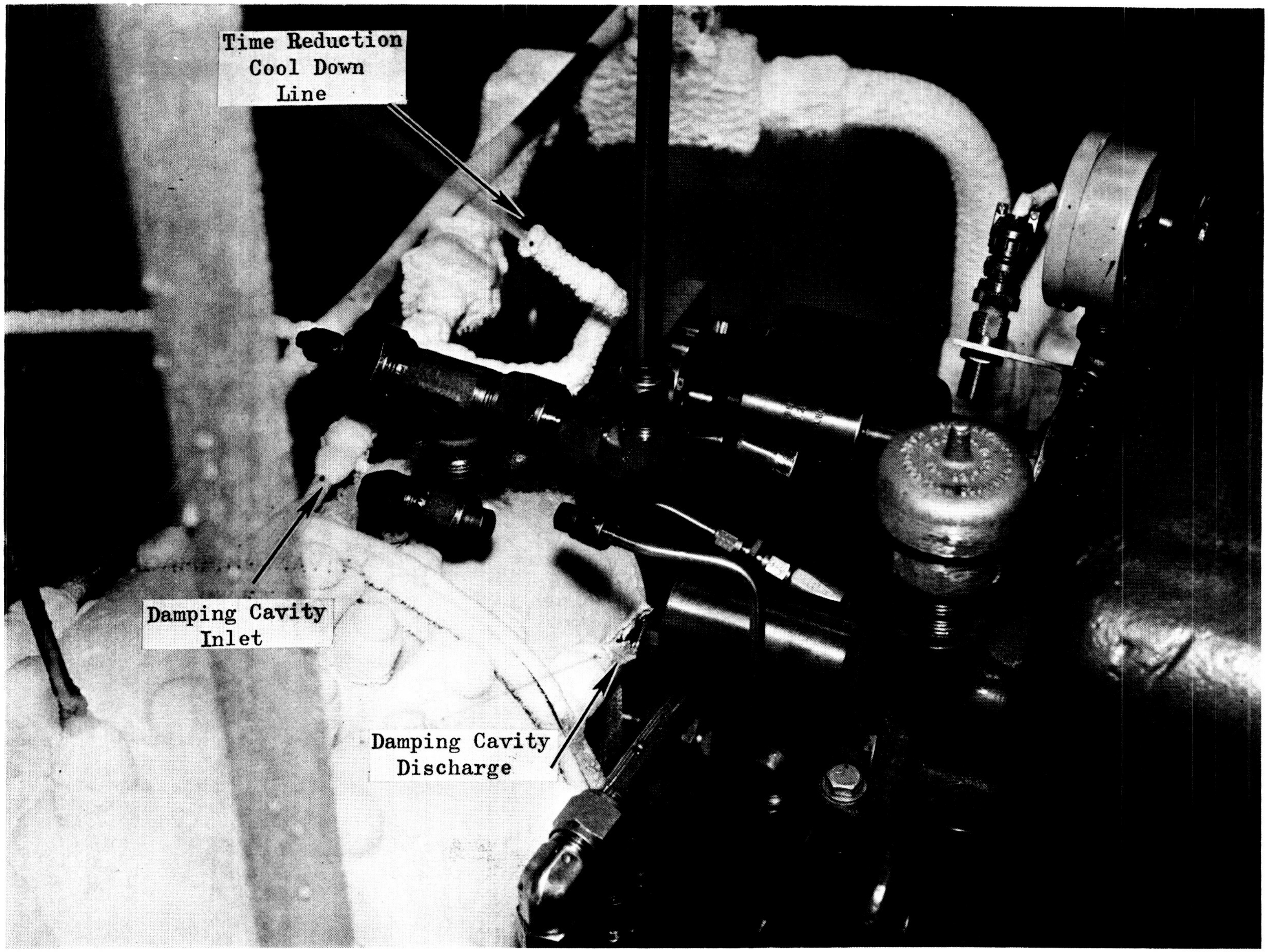

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Figure 28. Test Setup, Orifice Damped Seal (View 2)

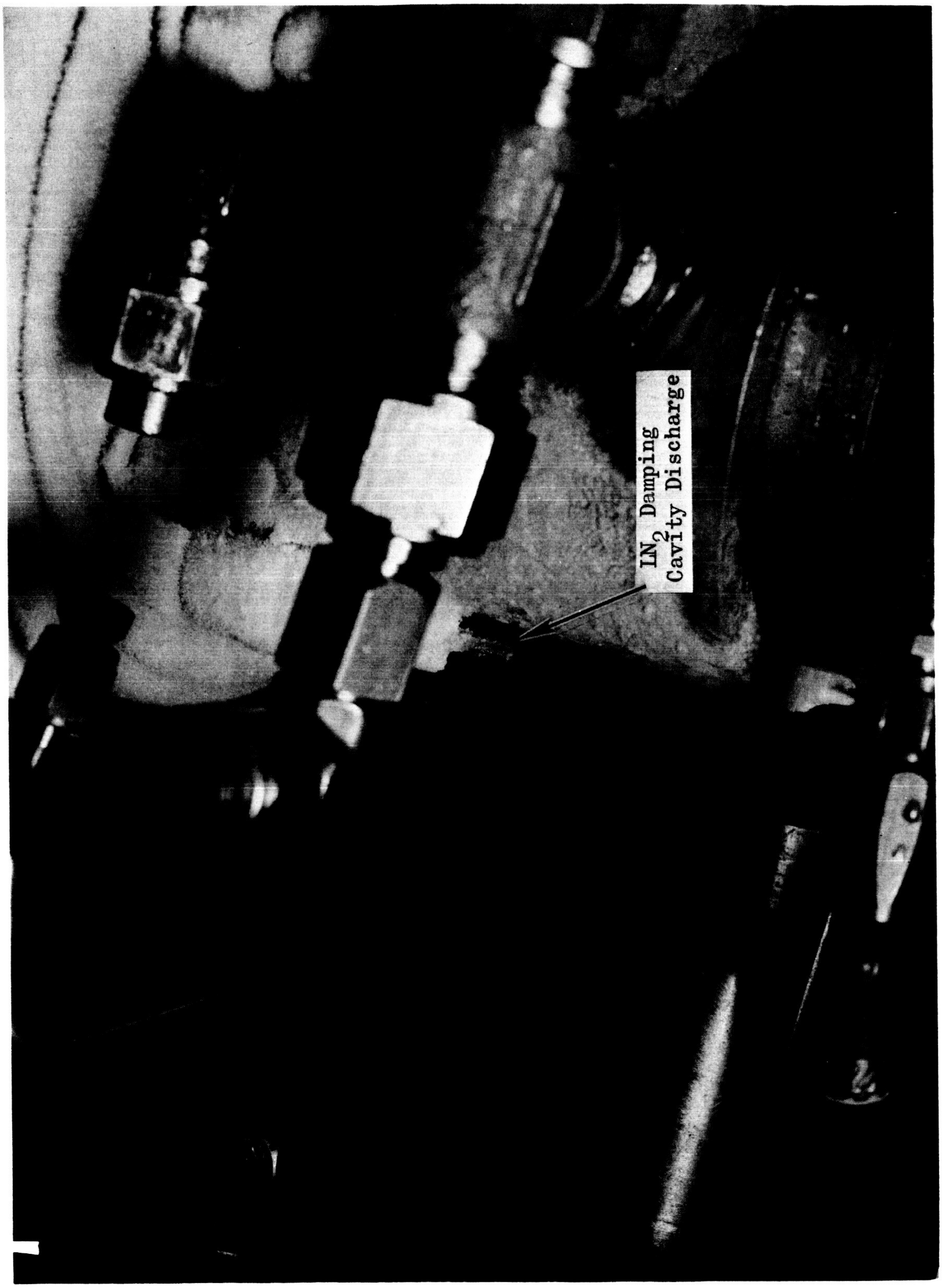

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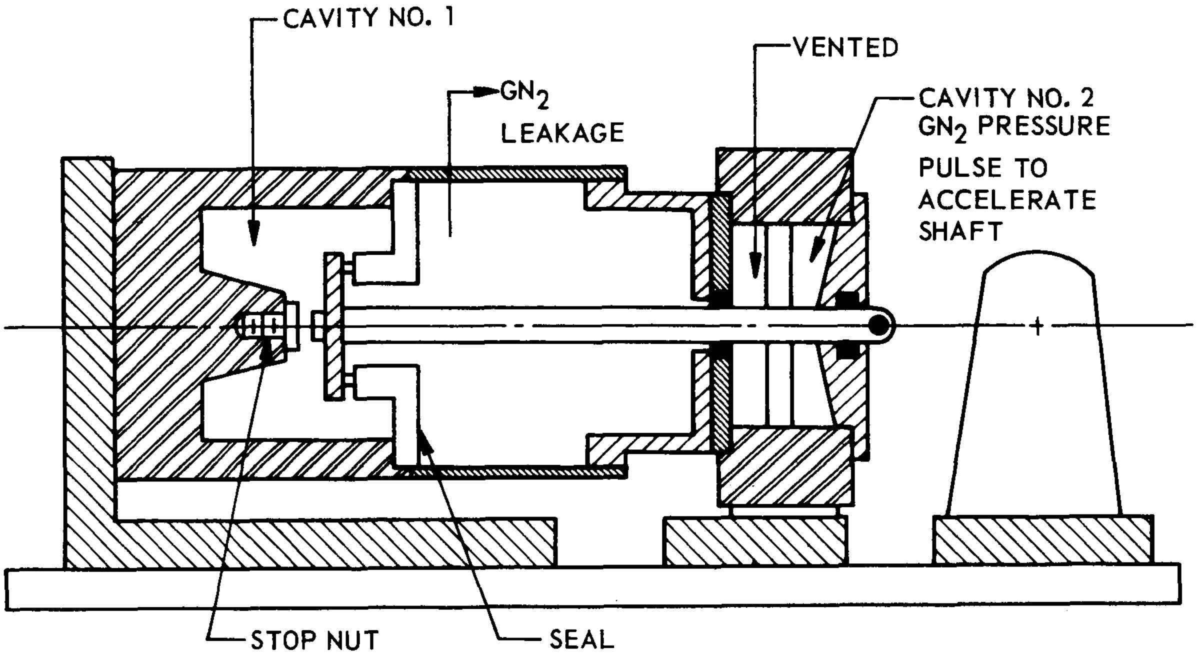
Figure 29. Test Setup, Orifice Damped Seal (View 3)

operated the eccentric crank to provide a shaft axial movement of 0.060 inch, thus compressing the bellows 0.030 inch and allowing the bellows to return 0.030 inch past the installed height. The motor speed was adjusted to maintain 30 cps for a period of 1,000,000 cycles. The test was completed with no mechanical difficulties.

Because of multiple leakage flow paths downstream of the bellows sealing face, a known leakage rate was not obtained. The housing does not accommodate a static seal capable of withstanding the 900 F environment. Asbestos cloth was used in place of the normally used rubber O-ring. Because of the porous nature of the cloth, it permitted the escape of GN_2 in the form of very small bubbles at a slow rate as indicated by the application of leak detection solution. The leakage can be described as very low and probably in a range of from 0.5 to 3.0 scfm as is normally the case. Posttest inspection of the seal was indicative of satisfactory performance. No residue was apparent on the seal to indicate a loss of NaK. The test results showed that the response of the bellows relative to shaft and mating ring displacement was not overdamped by the use of NaK. A mechanical cycling test would not normally show the effects of damping because the frequency is only 30 cps. Primarily, the mechanical cycling test is intended to demonstrate bellows integrity when the seal is exposed to a pressurized high-temperature system.

RECOVERY RATE TESTS

Following the mechanical cycling test of the orifice damped seal, a recovery rate test was conducted using the same seal containing the same volume of NaK. The test consisted of displacing the shaft and mating ring in a direction of decreasing bellows loading at a velocity greater than expected in advanced turbomachinery. Two linear transducers monitor displacement outputs, one mounted on the bellows sealing face and the other mounted on the shaft. With the eccentric crank removed, the method of shaft displacement (depicted in the following diagram) was accomplished by actuation of a pressure solenoid to produce a force in cavity No. 2.



To reach the highest practical shaft velocity with the illustrated picture, the pressure in cavity No. 1 was reduced to 10 psig while the pressure in cavity No. 2 was 250 psig. The elapsed time to travel the distance of 0.060 inch to the stop was found to be 10 milliseconds. The following calculation is based on the model shown in Fig. 21 and indicates that the shaft would have to travel the distance in less than 4.5 milliseconds before separation could occur.

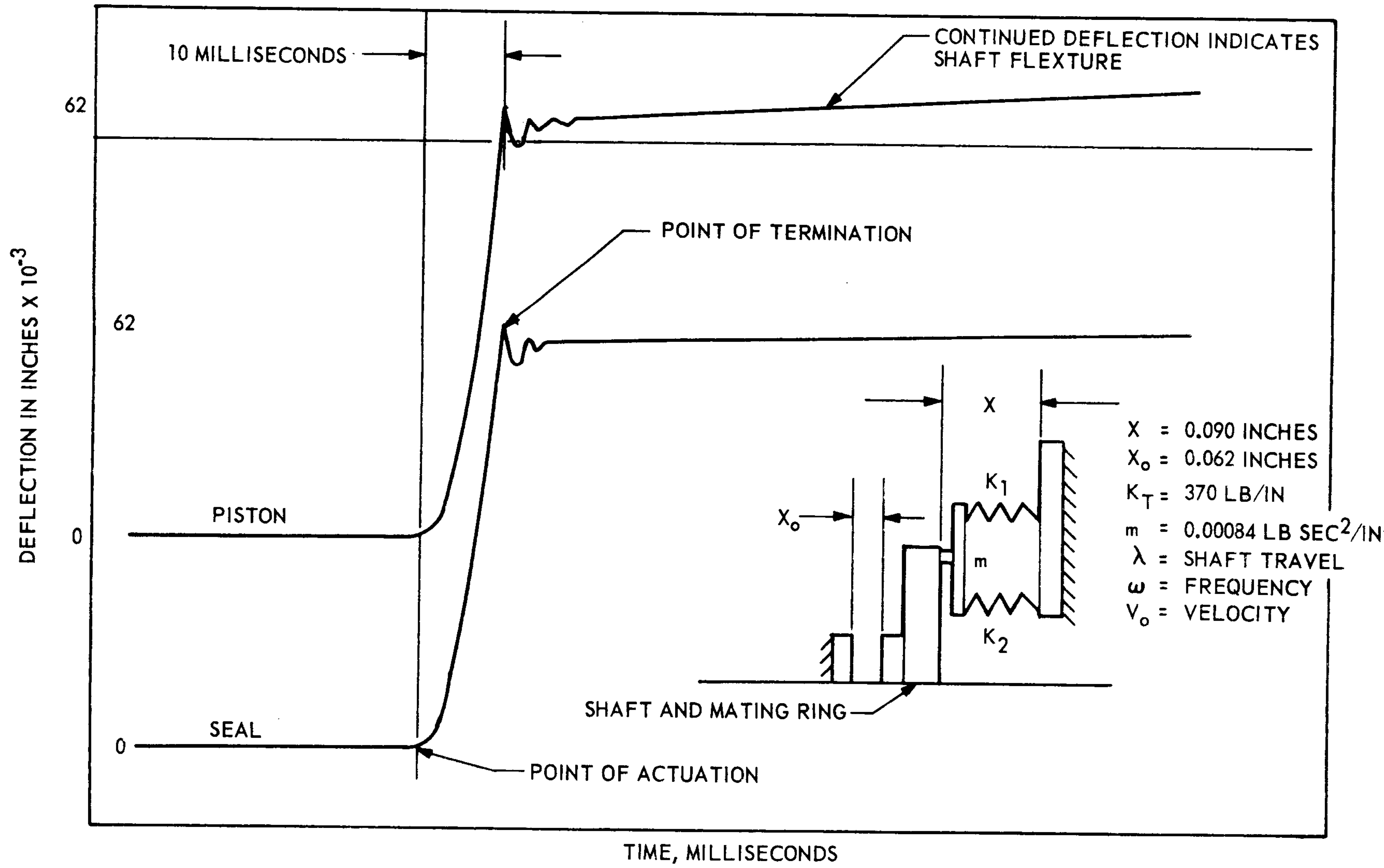


Figure 30. Recovery Rate Test Orifice Damped Seal (NaK Filled, S/N 6)

The spring mass system is represented by the equation

$$m \frac{d^2x}{dt^2} + Kx = 0$$

Kx is positive because the bellows is in compression

where

$$w = \sqrt{\frac{K}{m}}$$

w = natural frequency

k = spring rate

m = spring mass

The general solution is given as

$$x = A \text{ Sin } wt + B \text{ Cos } wt$$

where

A and B are arbitrary constants

At the compressed height (x) of 0.90 inch, the following initial conditions exist:

at $t = 0$ (1) $X_o = 0.062$ inch when $t = 0$ the Sin wt becomes 0

(2) $\frac{dx}{dt} = V_o = 0$ $\therefore$ $A = 0$ and $B = X_o$

then the solution is $\lambda = X_o$ $\cos wt = \frac{\lambda}{X_o} = 1$

$$\therefore t = \text{Cos}^{-1} \frac{1}{w} = \cos^{-1} \sqrt{\frac{m}{K}} \text{ so, } t = \sqrt{\frac{0.0074}{370}} = 4.5 \text{ milliseconds}$$

This time is based on a nondamped system. The test results show that the bellows are not overdamped at the conditions tested.

A recovery rate test was also conducted on piston damped seal S/N 3. The seal was installed at a compressed height of 0.090, and a mating ring displacement of 0.058 imposed which is 83 percent of the bellows free maximum travel height. With a ΔP of 240 psig, the mating ring and seal moved the 0.058 distance in less than 10 milliseconds without separation, and no increase in leakage was observed.

PRESSURE CYCLING TEST

An orifice damped seal was installed in the setup shown in Fig. 31 which was assembled from available hardware. The test consists of imposing oscillating LN_2 pressures on the primary seal bellows with the damping cavity filled with recirculating LN_2. The seal was installed at the mean design compression and remains at that compression unless influenced by the pulsating pressure as indicated by monitoring primary seal leakage.

The seal cavity and LN_2 accumulator bellows were pressurized to 200 psig, and the balance piston was pressurized to over 2000 psig to balance the force produced by the accumulator bellows assembly. The hydraulic actuator was energized by a servovalve to pulsate the LN_2 ± 50 psig. The response of the pressure transducer located in the seal cavity is shown in Fig. 32 for 3, 10, 20, and 30 cps. A pure sine wave was not apparent in the recorded data. The cause is due to a time lag in transducer sensing relative to the pressure actuation at the accumulator bellows. This condition is reflected at the transducer as a sine wave with a susperimposed harmonic stimulated by the accumulator bellows approaching a resonant condition. A further increase in frequency (from 10 to 30 cps) causes a greater change in amplitude. This increased amplitude was apparent during the test by a gross movement of the entire test apparatus.

① Seal Pressure: 200 + 50 psig at 100 cps

② Seal Temperature: -323 F

③ Seal Leakage: 0 to 1000 scims

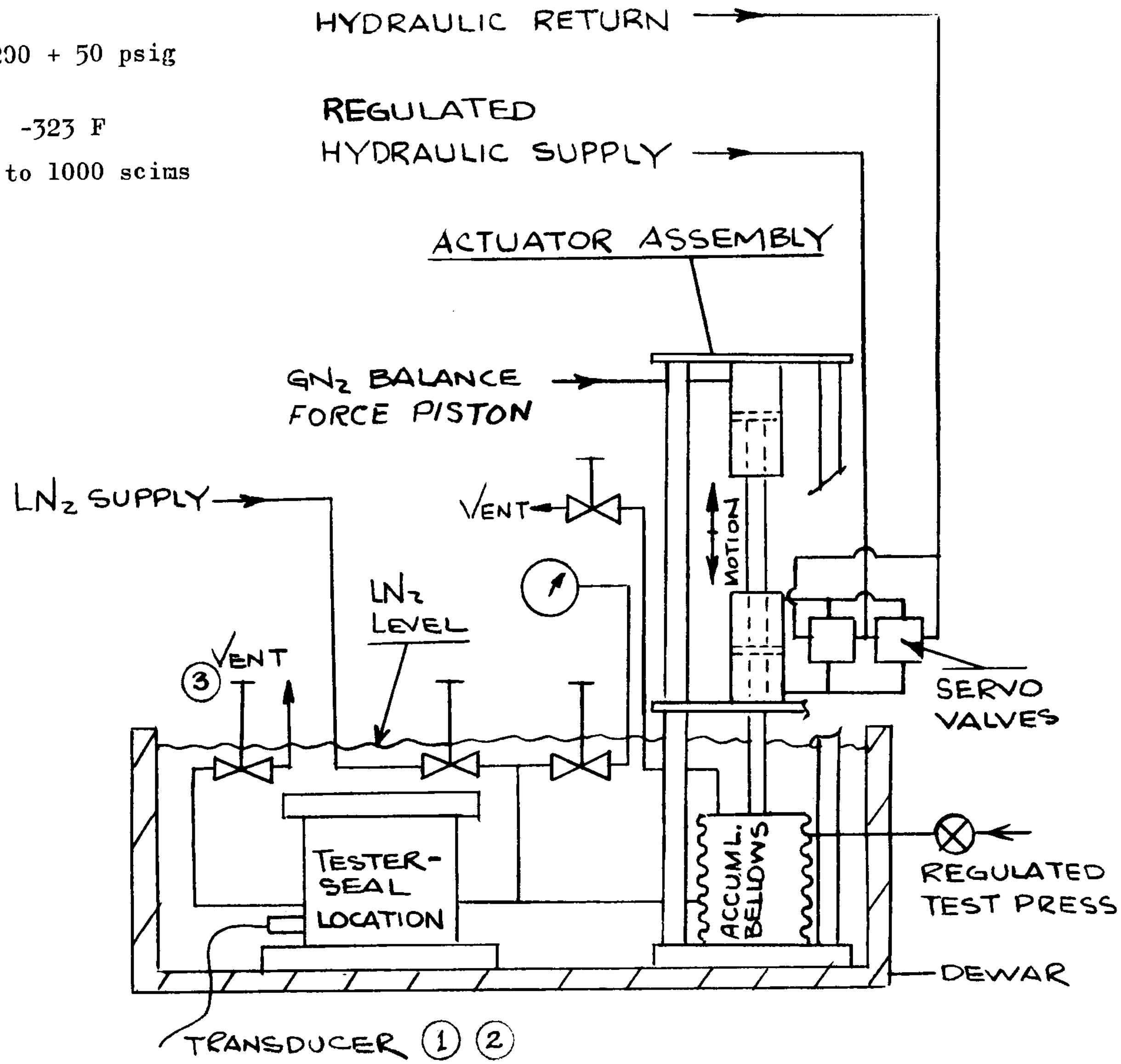


Figure 31. Pressure Cycling Test

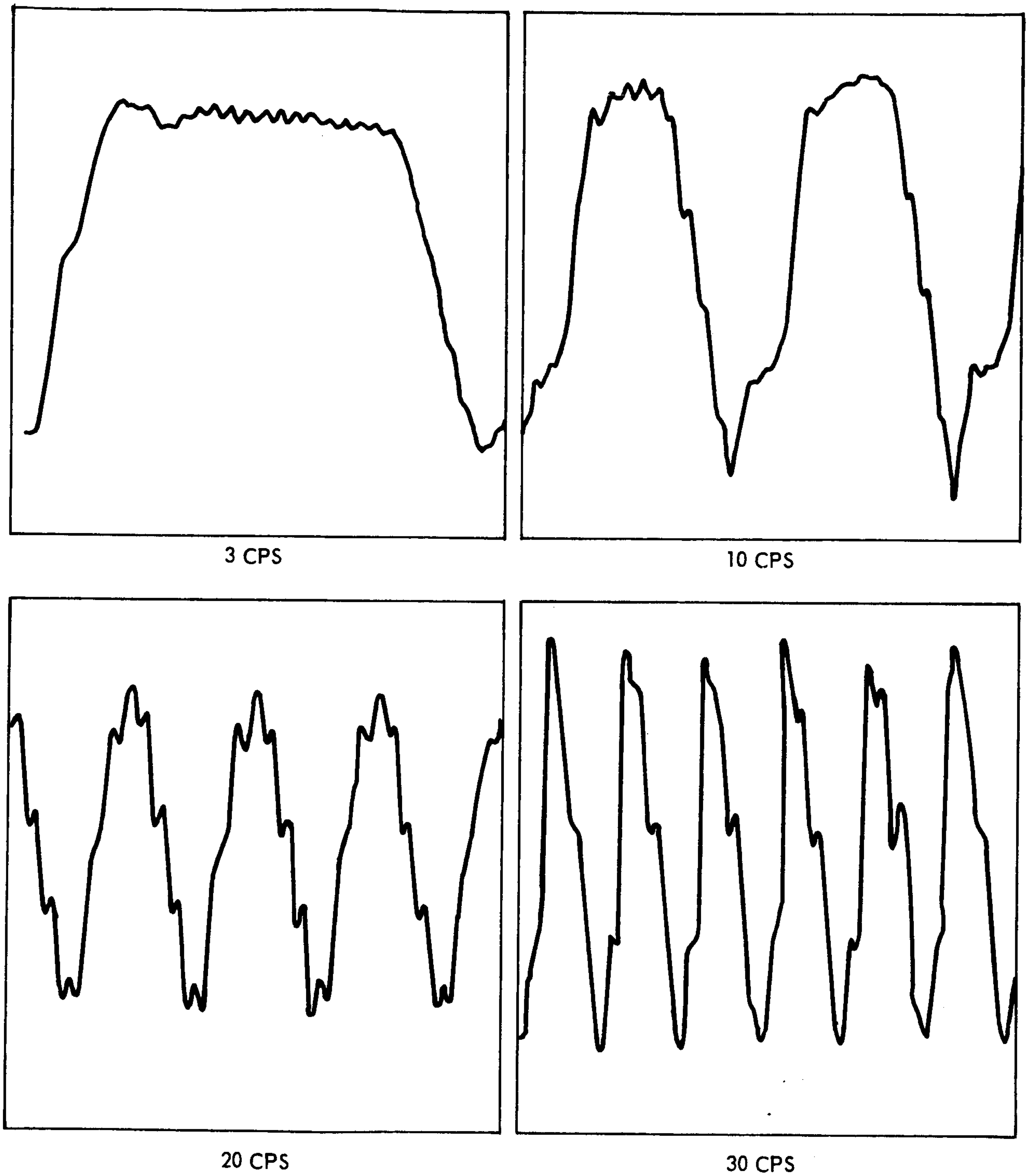


Figure 32. Pressure Cycling Test Data

Attempts to modify the system were considered; however, after 180,000 cycles (90 percent at 30 cps) excessive leakage was noted at the gas flow-rates and resulted in test shutdown. Disassembly of the tester and inspection of the bellows seal indicated a bellows rupture at the seond ID convolution approximately 1 inch long.

Mode of failure was ascertained to be fatigue of the convolution weld bead in the heat affected zone. The test duration of 180,000 cycles is the number of cycles produced by the actuator piston. The number of cycles imposed on the primary seal bellows is approximately 3 to 4 times 180,000 cycles, as can be noted in Fig. 32. The amplitude of the additional cycles is in the order of 10 psig.

The recirculating LN_2 used as the damping fluid was subject to a volume/density change which indicates it may be necessary to limit use of this design to fluids that can be contained and sealed in the damping cavity.

TOTAL FACE LOADING TEST

A series of total load tests were conducted on piston damped seal S/N 5 using the apparatus shown in Fig. 33 . The tester is used in conjunction with an Instron machine which controls the shaft movement and records the load exerted on the load cell. A remote system to supply and monitor the test pressure is also used. The Instron machine and tester are shown in Fig. 34. The recording equipment is located outside the test cell.

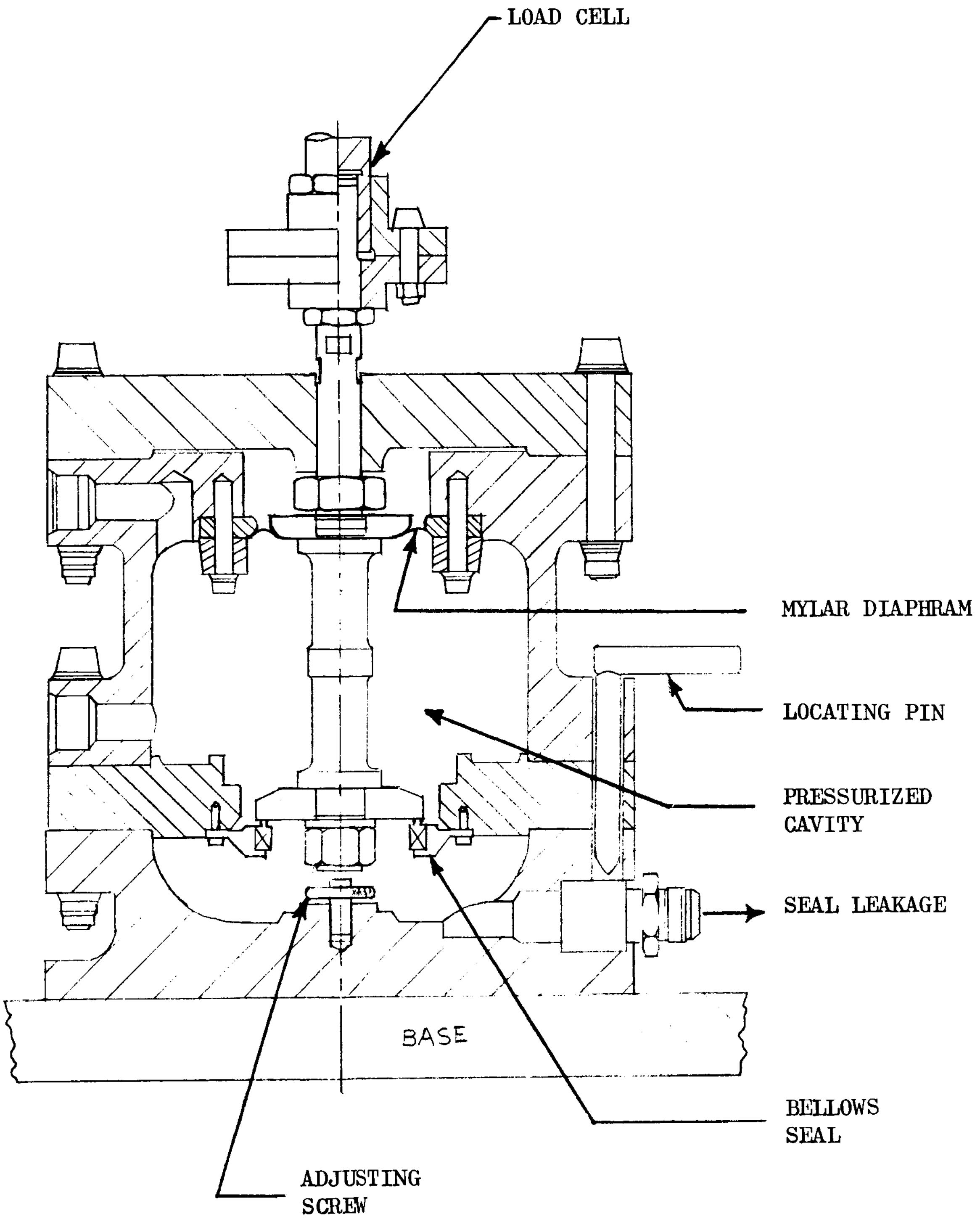


Figure 33. Total Face Load Tester

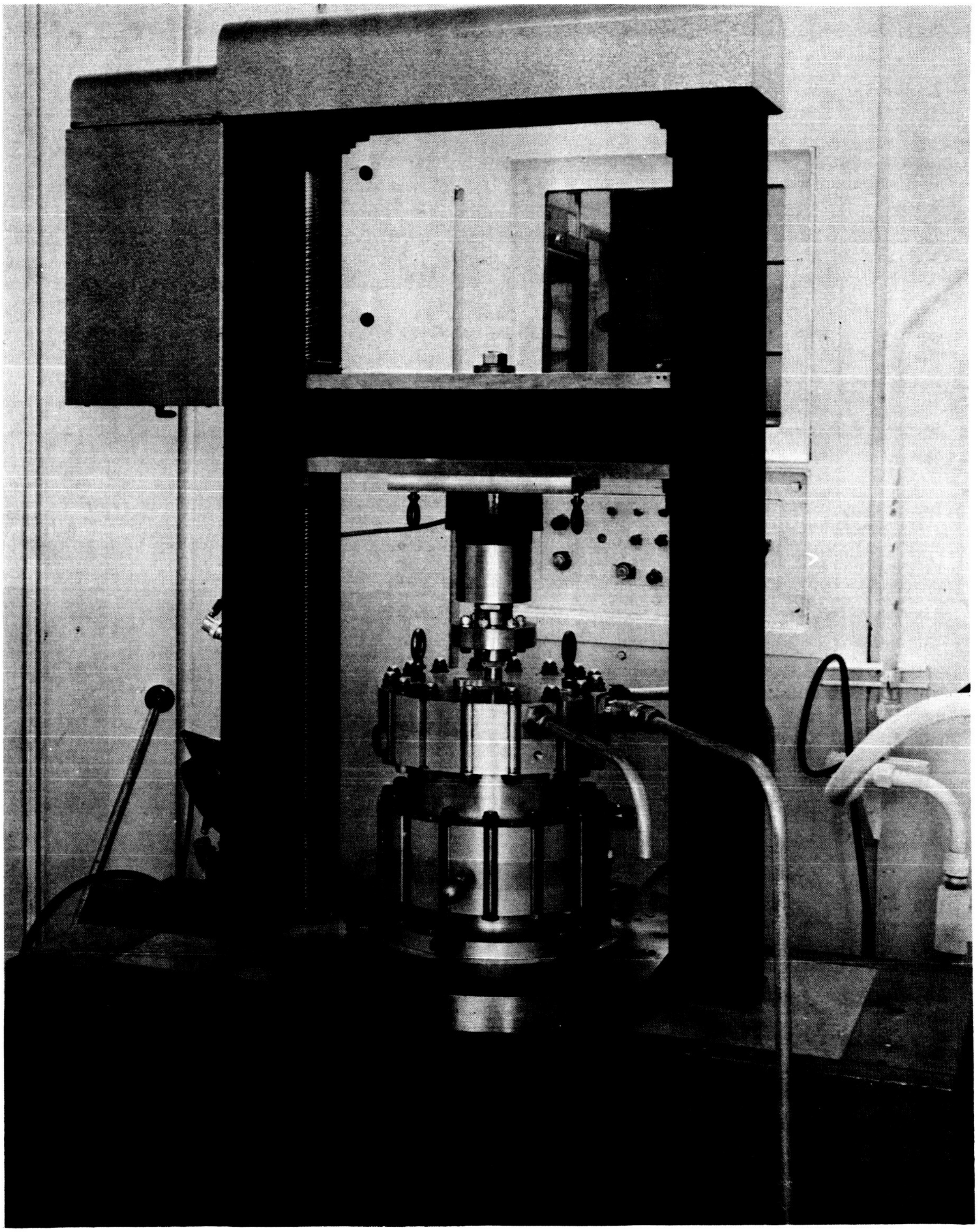


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Figure 34. Total Load Test Setup

As discussed in the design section on the piston damped seal, the mechanical characteristics involved in face seal operation must be known to establish limits of design parameters. The primary factors concerned in bellows design which predict operational characteristics can be determined through total load tests are as follows:

1. Spring force of the bellows through the desired operating range
2. Effective hydraulic area
3. The change of the bellows effective diameter caused by deformation of the bellows plates when varying the operational pressure
4. The change of seal face unit loading with respect to a change in compressed length or a change in pressure
5. Seal leakage to test the adequacy of the pressure balance

The test conducted on the piston damped seal was accomplished with existing hardware with the exception of the Mylar diaphragm used in the tester to maintain a static pressure environment. Prior to installation of a seal, the diaphragm was calibrated through the planned operating range of the bellows to account for the effective area change of the diaphragm. Figure 35 represents the plotted total load data for S/N 5.

The plot of bellows effective diameter (Fig. 36) is calculated data based on the total load data of Fig. 35 . As can be noted some of the load values oscillate. These oscillations are assumed to be the result of bellows plate deformation under pressure causing plates to touch and share the load non-uniformly. Under normal operation, effective diameter change is only 0.005 inch at the design operating point of 200 psig.

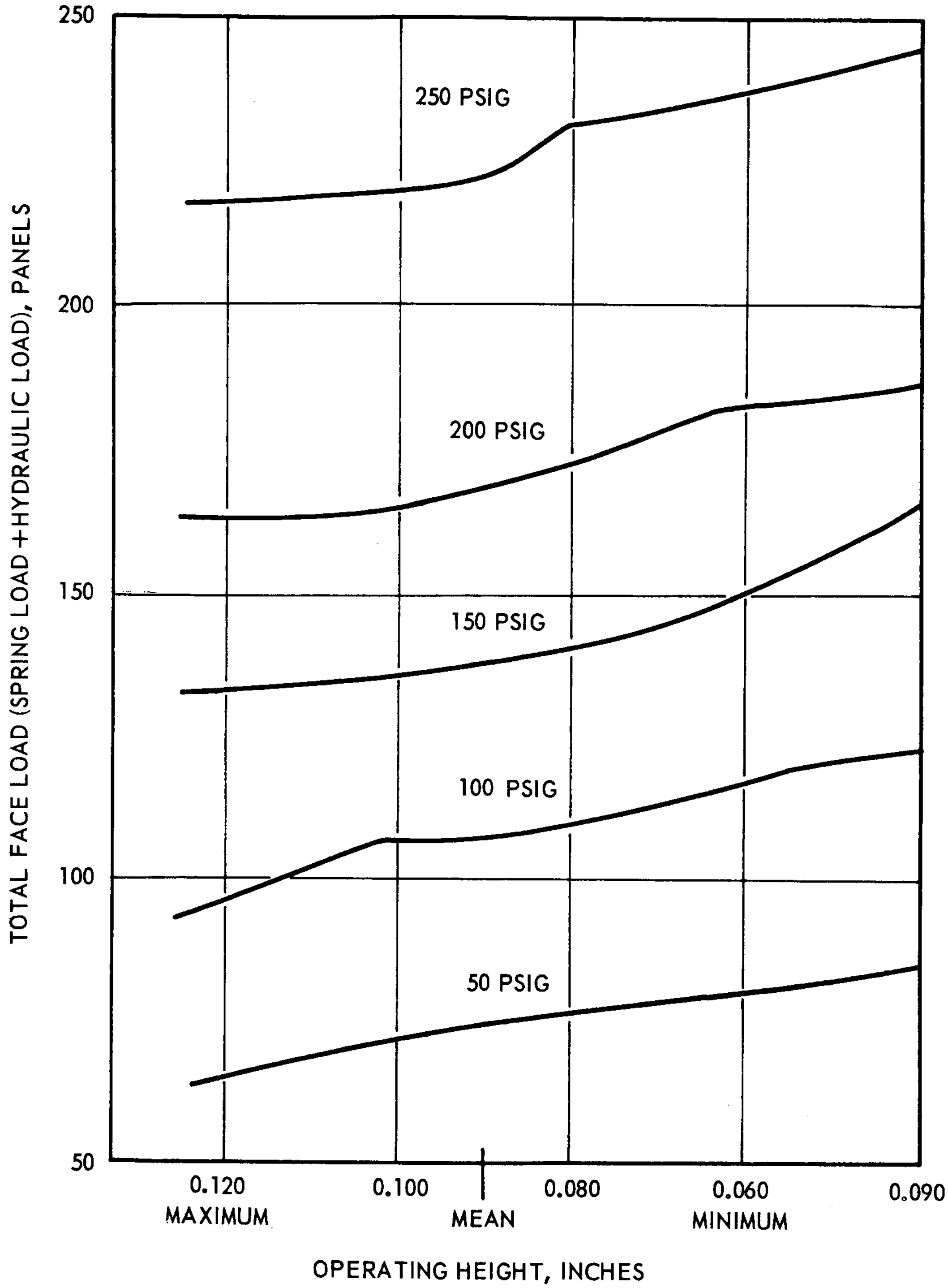

Figure 35. Piston Damped Seal S/N 5 (Data Compares Closely to Vendor Data)

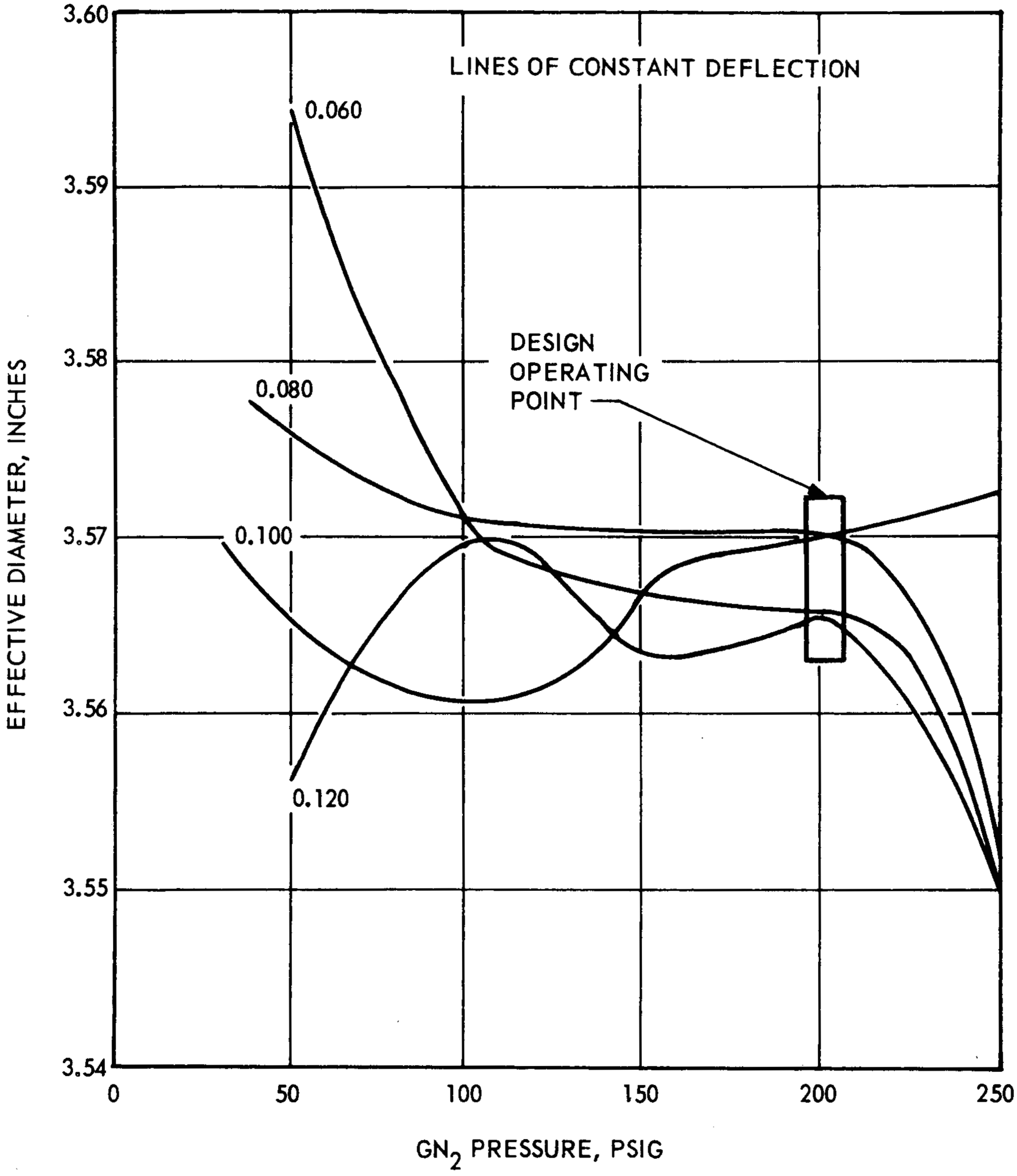

Figure 36. Effective Diameters Calculated Based on Total Load Data Minus Spring Load

VIBRATION TESTS (PISTON SEAL)

A series of vibration tests was conducted on a piston damped seal with and without a damping piston to observe seal face response at resonance. The seal was installed without a piston at the mean operating height in which the bellows is compressed 0.110 inch. A test housing encloses the seal to provide a volume for a GN_2 or LN_2 environment at the bellows OD. Displacement is monitored by two linear transducers, one attached to the bellows carrier and one mounted on the test housing to monitor relative movement.

With no piston installed, two tests were run at 2 g and 5 g. A frequency sweep to 2000 cps showed three pointsof resonance (606, 1400, and 1800 cps). Both transducers indicated a maximum amplitude of 250 microinches. With a piston installed (0.002 clearance) the tests were repeated. The results indicated no change in amplitude. The environment was gaseous nitrogen heated to 900 F and pressurized to 200 psia. Results of tests support the previous conclusion that damping will not occur with this system at small displacement amplitudes.

To increase the possibility of seal face liftoff, the compression load of the bellows was decreased beyond normal operation. With the bellows compressed 0.010 inches, the seal was subjected to the same frequency sweep but only the first indication of resonance was considered. The seal was exposed only to normal room atmosphere; at approximately 600 cps, liftoff occurred with a maximum amplitude of 0.002 inch. The test was repeated using LN_2 to note any change in amplitude caused by a fluid condition pressurized to 200 psig.

The resonant frequency and maximum amplitude decreased by a very small amount under LN_2 environmental conditions and indiated that little damping existed with the displacements involved and the type of fluid used. The quality or liquid/vapor fraction of LN_2 used may be questionable because considerable leakage occurred at the resonant frequencies. This amount of gas leakage from the liquid supply could cause a two-phase fluid condition.

VIBRATION TESTS (PARTICLE SEAL)

Initially, a series of tests were conducted to document the performance of the particle damped seal to compare the results with the beam testing described previously of the particle damped seal design and to establish the most effective level of fill of the particle containers. One piezoelectric accelerometer was bonded to the seal face with its exciting axis parallel to the seal's longitudinal axis. A control accelerometer was installed on the vibration table to monitor input excitation. The test setup used is shown in Fig. 37. No special environment was used nor was the bellows restrained. The motion of the seal was free axial movement responding to an input acceleration of 0.5 g. The molybdenum particles used to examine the damping effect were chosen based on previous beam tests which show that particles having a -325 sieve size rating give the best results. This sieve size includes particles of 44 microns and below. A random sample is shown in Fig. 38. Close examination shows the particles to be spherical in shape and having a relatively smooth surface.

The tests were conducted with an exciting force held constant (0.5 g peak) and the frequency varied from 20 cps to 1 kilocycle. Relative damping was determined by measuring the bandwidth of the resonant peak at the half-power

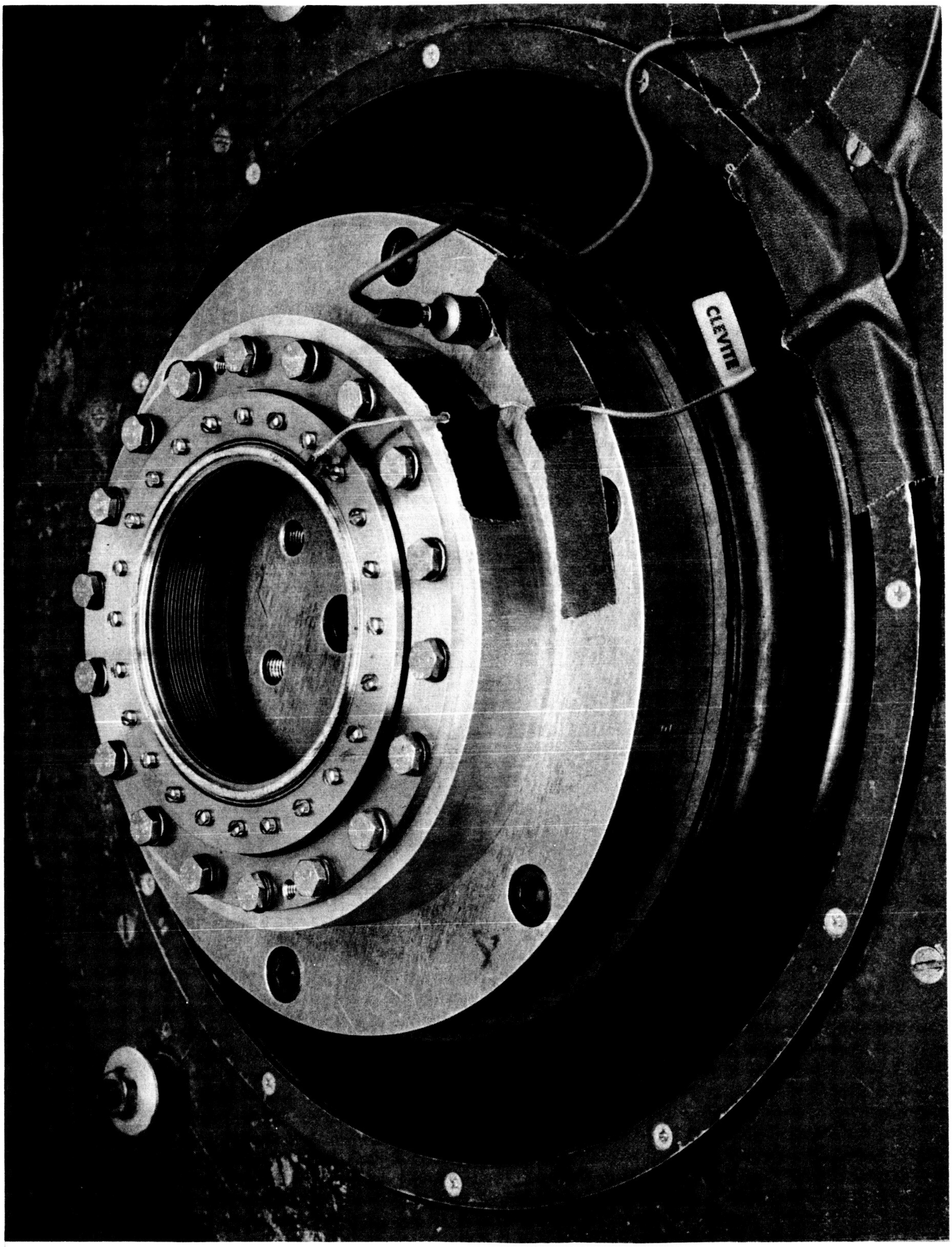

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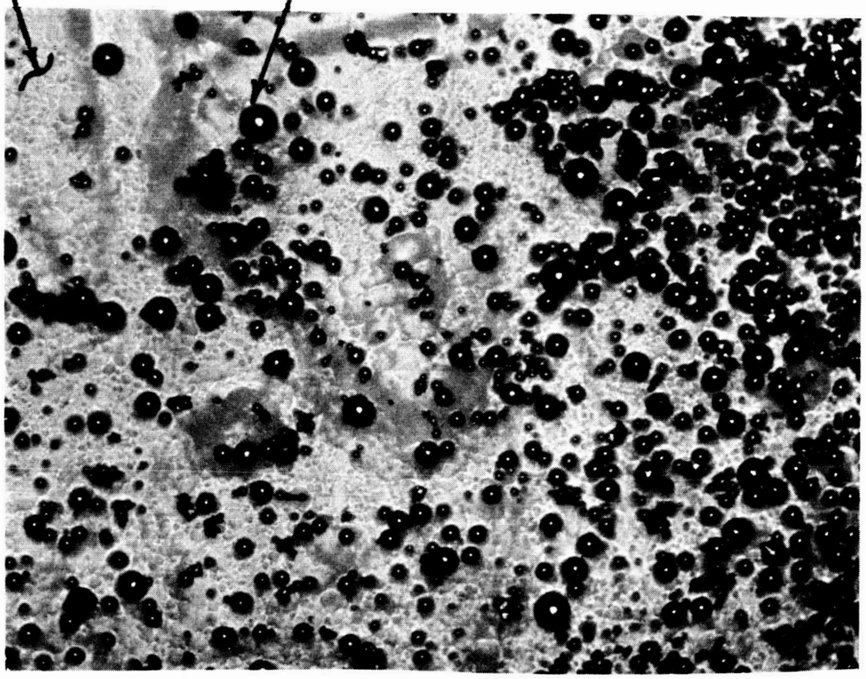
Figure 37. Vibration Test Setup, Particle Damping Seal

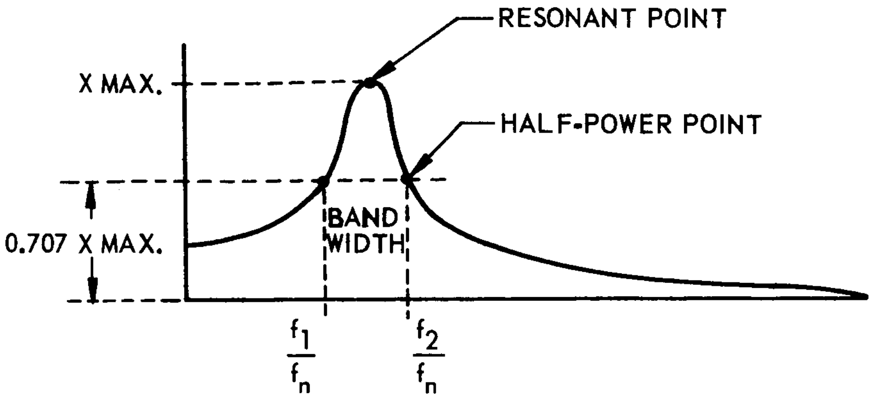


150 X

Largest Particle is
44 Microns in Size

Figure 38. Molybdenum Spherical Powder

points in terms of normalized frequency. The diagram below illustrates the damping measurement.



As the bandwidth increases, relative damping also increases. The formula used to compute relative damping for this case is:

$$\Delta f = \text{Relative damping} = \pi \left(\frac{f_2}{f_n} - \frac{f_2}{f_n} \right)$$

where

f_n = resonant frequency

f_1 = lower half-power frequency

f_2 = upper half-power frequency

In addition to damping, the frequency at resonance and the amplification at resonance were measured and recorded. The amplification is defined as the ratio of acceleration into the mass to the acceleration out, and is another term used to define effective damping.

$$\text{Amplification} = \frac{G \text{ out}}{G \text{ in}} \quad \text{or} \quad \frac{A_o}{A_i}$$

The following are the tabulated test results:

TABLE 7

TEST DATA, BELLOWS UNRESTRAINED

Test No.	Capsule Condition	f_1	f_n	f_2	Δf	A_o/A_i	G in	G out
1	Empty	75.5	75.9	76.3	0.033	108	0.5	54
2	1/4 Full	71.1	72.5	72.6	0.065	51	0.5	25.5
3	1/2 Full	66.5	67.7	68.3	0.0835	43	0.5	21.5
4	3/4 Full	62.4	63.5	64.5	0.1039	35.2	0.5	17.6
5	Full	60.0	60.6	60.7	0.0364	93	0.5	46.5

The test results for two cases, empty and the most effective, show that a reduction in amplification of 66 percent occurs from empty to 3/4 full.

A series of tests were conducted to establish the amount of damping available with the spring mass system restrained as in actual operation. The bellows was compressed 0.085 inch by a disc mounted as an integral part of the vibration table, therefore imposing a constant load on the bellows.

Tests were conducted in a frequency sweep range from 30 cps to 2 kilocycles at vibration input levels of 1, 5, 10, and 20 g rms with the particle holders both empty and 3/4 full. A short test was also run at 3/8 full to verify the particle holder optimum filling level.

The accelerometer data was recorded on log paper plotting the frequency vs bellows response (g rms). As a result of loading the bellows, several points of resonance occurred at different frequencies. Damping could not be defined in terms of the fundamental frequency and the half-power points as was the case in the unrestrained bellows tests; therefore, maximum output response (g rms) was used and damping referred to as the ratio of the two outputs, empty and 3/4 full. Table 8 presents the data for the tests showing damping as:

$$\Delta = \text{damping} = \frac{\text{g rms undamped}}{\text{g rms damped}} \quad \text{with constant input}$$

TABLE 8

TEST DATA, BELLOWS RESTRAINED

Input G_I	Maximum Response G_o	Frequency at Maximum Response	Damping Δ	Particles
1	28	1173	7.4	0
1	3.8	906		3/4 full
5	72	1039	2.08	0
5	34.8	1065		3/4 full
10	118	1021	2.2	0
10	53	1066		3/4 full
20	(damping not defined for 20 g case caused by bellows failure)			

A typical response plot is represented in Fig. 39 and 40 for the 10 g case.

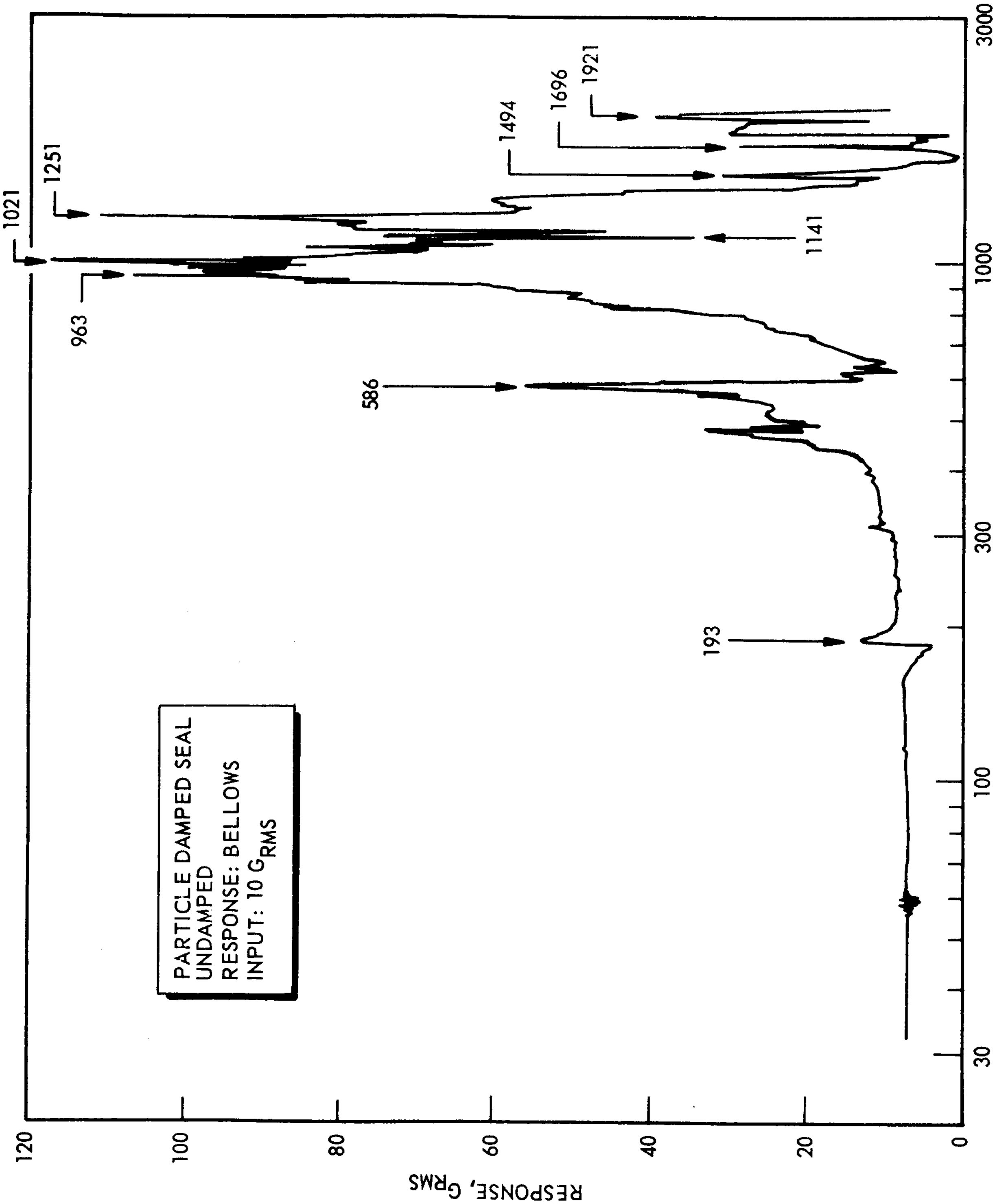


Figure 39. Bellows Response, 10 g, Particle Undamped

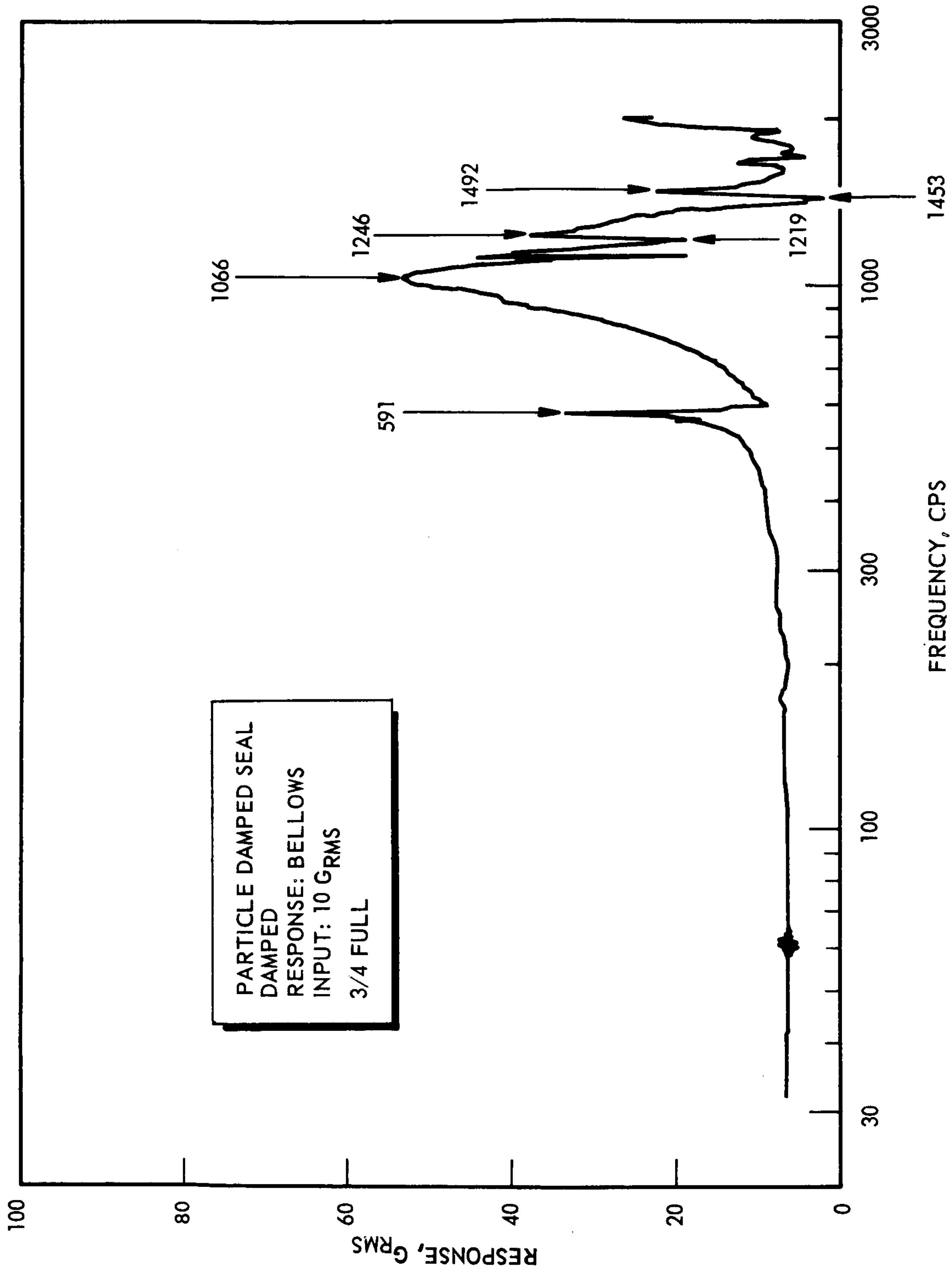


Figure 40. Bellows Response, 10 g, Particle Damped

The 20 g data shown in Fig. 41 and 42 indicated an unreasonable response compared to the lower level inputs of 5 and 10 g. Disassembly of the test apparatus and inspection of the seal revealed a severe bellows rupture at the ID of the second convolution from the rear bellows carrier. The observed rupture is indicative of a fatigue failure. The occurrence of the failure shows that the bellows life for this case was in the range of 35- to 40-million cycles and represents the total of the testing presented in Table 8 , and includes an average frequency of 1000 cps of which 10 percent of that duration was at resonant conditions. As the vibration test continued, the data became more erratic indicating a pronounced failure. The extent of the rupture was observed to be 75 percent of the welded convolution.

To establish a comparison between conventional friction and particle damping, a test was conducted on an experimental frictional damped seal which has been used in the J-2 Mark 15 oxidizer turbopump. The turbopump test results indicated a very low carbon wear rate with low leakage. Wave springs loaded against both the carbon carrier and bellows convolutions supply the vibration damping medium.

The response of the bellows is shown in Fig. 43 at the 10 g level input, the output is 35 g at a maximum amplitude at 1900 cps.

Comparing this data with the 10 g input particle damped seal data, the following information is apparent:

1. With the same vibration input, the output of the particle seal is 50 g vs 35 g for the frictional damped seal.
2. While the damping characteristics appear to be better for the frictional seal, it should be noted that the spring mass of the particle seal is approximately 10 times as great as that of the conventional seal, and therefore the damping required is correspondingly greater.

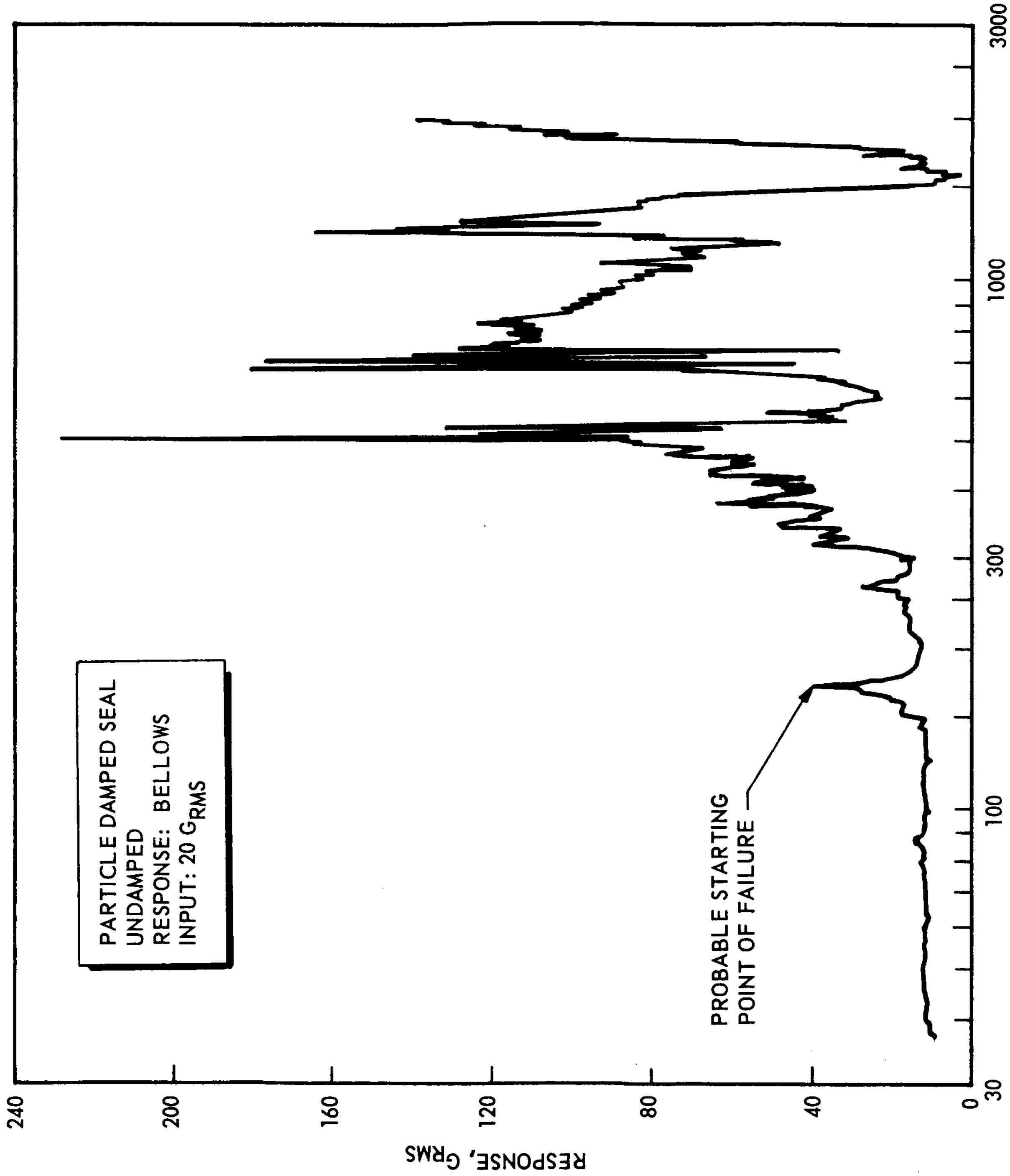


Figure 41. Bellows Response, 20 g, Particle Undamped

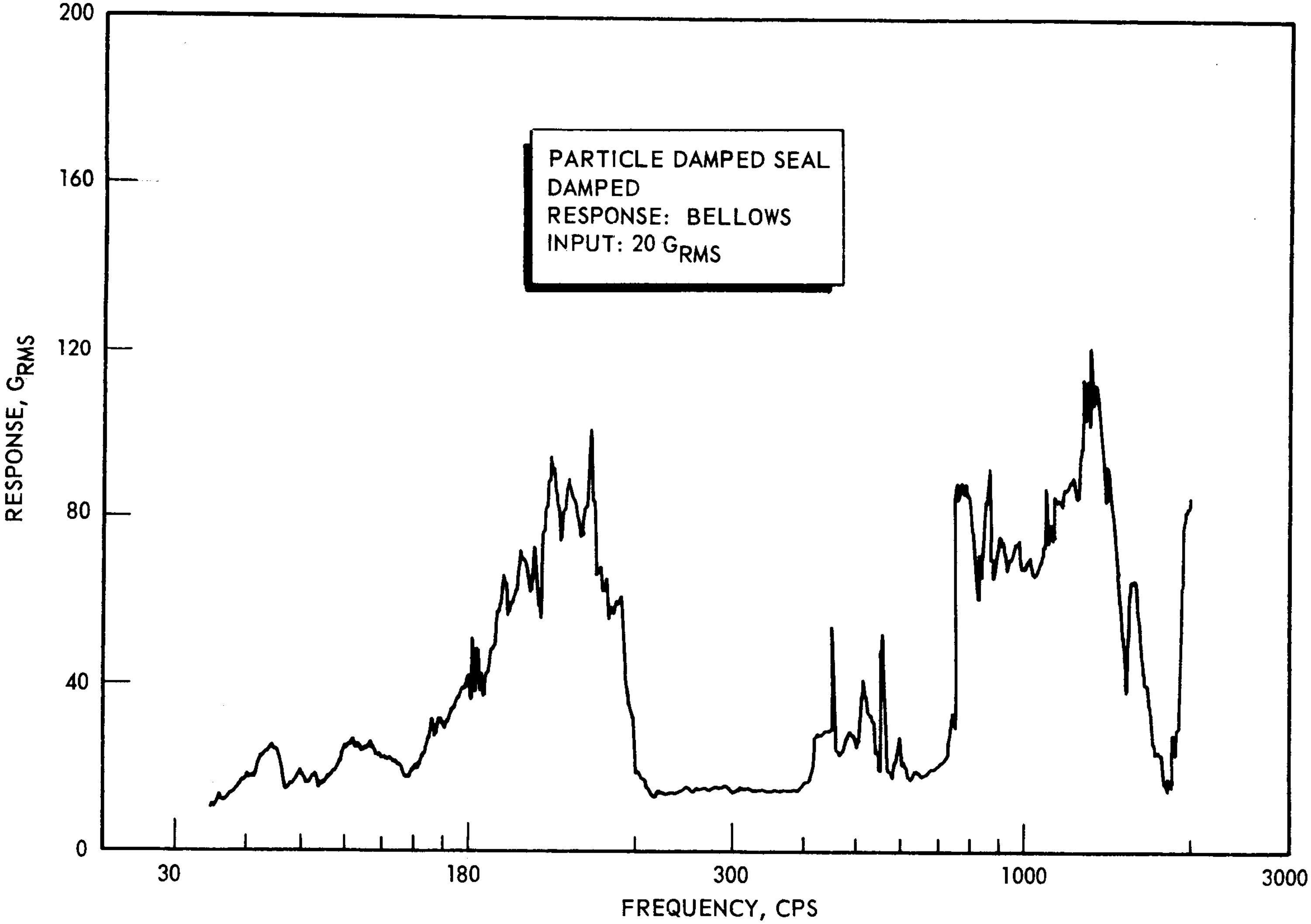


Figure 42. Bellows Response, 20 g, Particle Damped

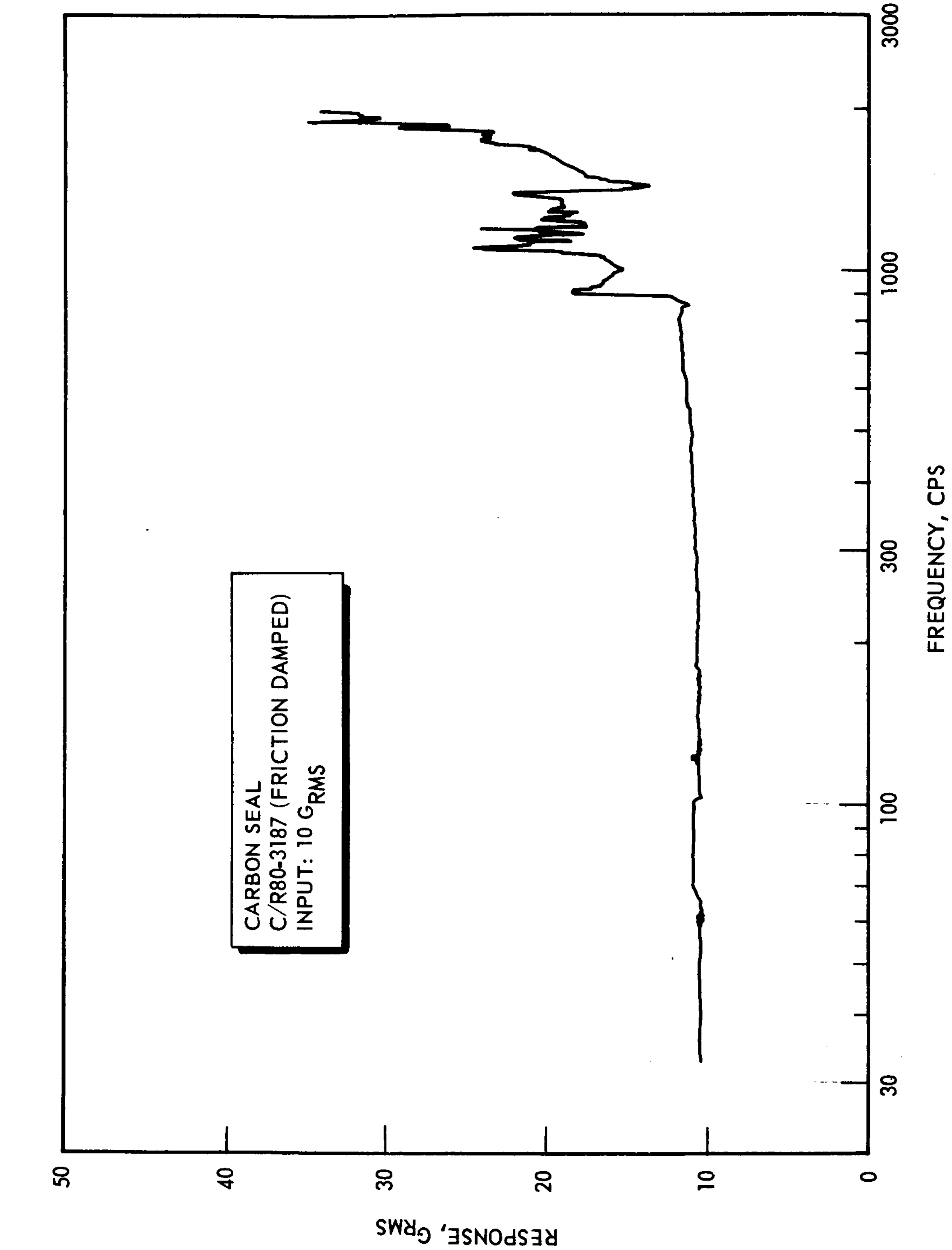


Figure 43. Bellows Response, 10 g, Friction Damped

The damping effect obtained by use of the particles is described in Table 8 and indicates a reduction in damping from 1 to 5 g input. A further increase in input to 10 g shows no relative difference in damping between 5 and 10 g, which may indicate a saturation point for the system tested. This condition makes apparent the need for further consideration in design of the particle containers to absorb additional energy. The results of the test data initiated the consideration of an improved design which incorporates stationary flexible baffles in the containers to respond to particle movement. Previous tests also indicate a greater damping effect is obtained when a greater number of particles are used. This indication is also being considered. The following is a program which is tentatively recommended to advance the particle damping concept and accomplish the above task.

Design

The use of the cylinders containing the particles in the current design requires an additional approach to increase the damping potential as indicated by the vibration test results. A preliminary design currently in process incorporates a method to include more particles with the containers having stationary flexible baffles responding to particle movement. The required amount of damping for successful operation of a turbopump bellows seal is not known at this time, nor is the principal mode of vibration identified. Although only the axial mode was tested in this program, the torsional mode may predominate in turbopump testing and could require greater emphasis. For this reason, and to observe seal performance, a series of tests simulating turbopump operation are recommended. This requires the addition of a carbon seal face to allow rotation.

Six seals are suggested for fabrication: three each of the seals designed to incorporate dimensions obtaining two different pressure balance ratios and two different bellows spring rates.

Dynamic Testing

The tests will be conducted at the Santa Susana Field Laboratory (SSFL) California and will use a modified J-2 Mark 15-F seal tester using pressurized LO_2 as the test fluid. Facilities include the necessary instrumentation to measure seal leakage, imposed vibration, fluid temperature, pressure, and shaft speed.

Generally, the test conditions recommened are:

1. Shaft speeds up to 20,000 rpm
2. LO_2 temperature at bellows OD, -323 F
3. LO_2 pressures up to 250 psig

APPENDIX

SEAL CONCEPTS CONSIDERED FOR EVALUATION

A description of the seals considered for evaluation is reported in the following summary.

LIQUID METAL SEAL

The liquid metal seal (Fig. 44) utilizes a liquid metal barrier to retain the sealed fluid. A ring of liquid metal is forced against the seal housing with sufficient centrifugal force to overcome the effect of the sealed pressure acting on the liquid metal in the opposite direction. This seal was selected for further study because it offers zero leakage possibilities. Problems of storing the heavy fluid, starting, and the stability of the interface must be resolved before the concept is suitable for rocket engine applications.

MOLDED SECONDARY SEAL

The molded secondary seal (Fig. 45) consists of a standard type nosepiece mated to a coil spring with a molded cover. The molded cover must be pliable and elastic and in addition, must be compatible with and provide acceptable sealing of the fluid. Because of the limited temperature capability (both hot and cold) of presently available materials,this seal concept was eliminated for further consideration.

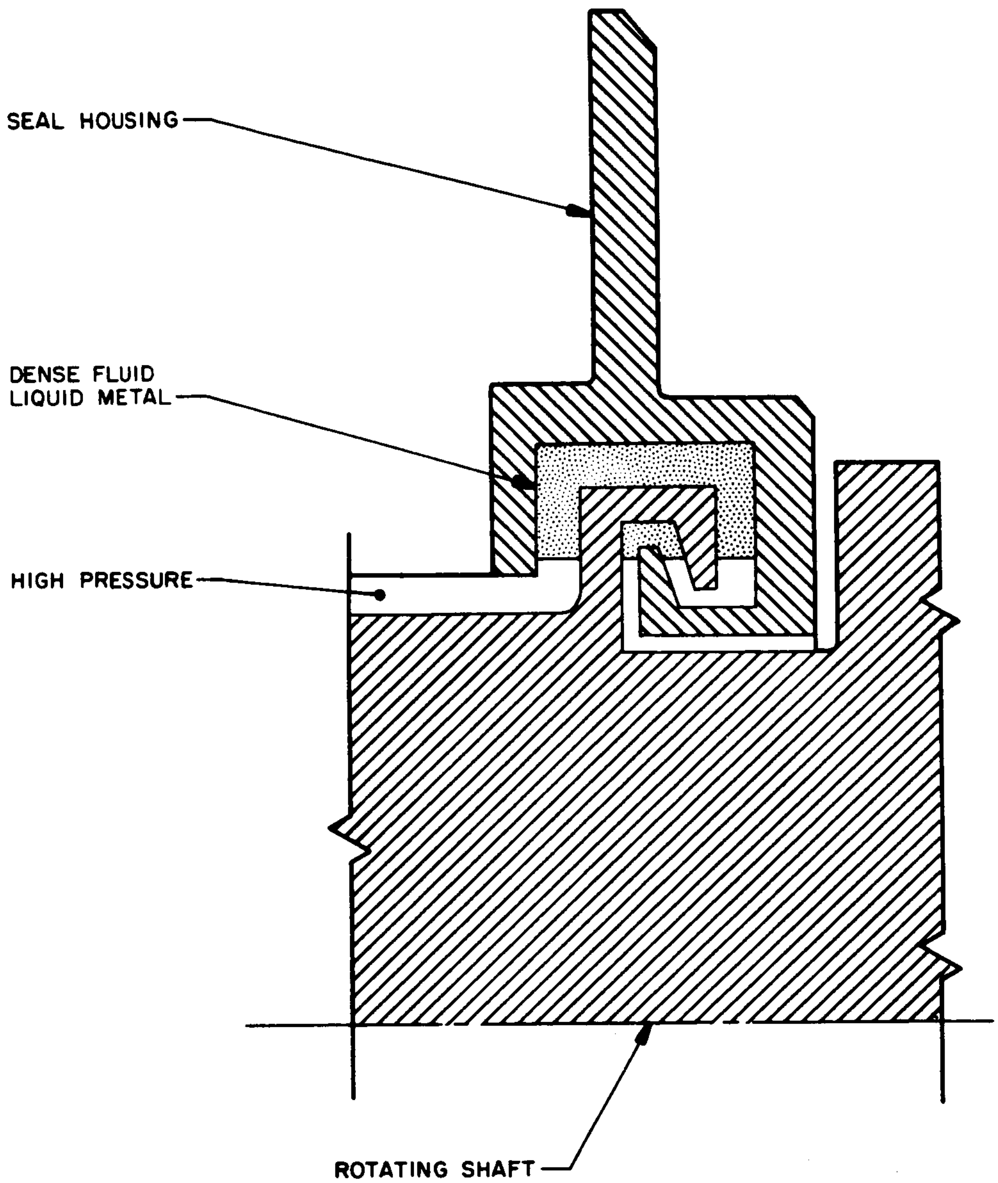


Figure 44. Liquid Metal Seal

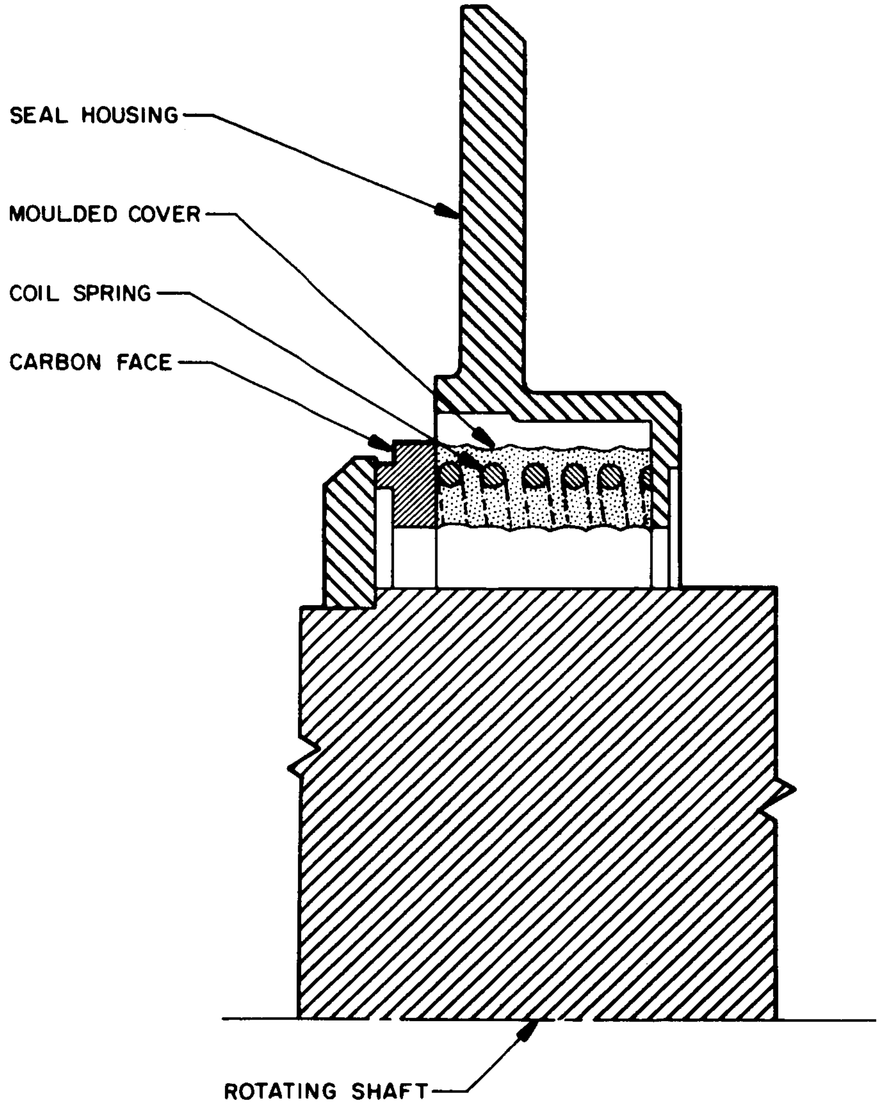


Figure 45. Molded Secondary Seal

HEATED LABYRINTH SEAL

The heat labyrinth seal (Fig. 46) is a two-stage, labyrinth-type seal consisting of a first stage to effect a pressure drop with liquid (cryogenic) flow, and a second stage to effect the final drop with gas flow. Between the two stages a heat source would ensure the liquid would be flashed to a gaseous state and its temperature increased. This seal is actually a controlled leakage device and a fairly high power consumption would be necessary to effectively vaporize the liquid. For these reasons the seal was dropped from further consideration.

SHAFT PUMP SEAL

The shaft pump seal (Fig.47) incorporates a spiral lip on the shaft to provide a pumping action to balance the leakage flow. The base of the housing would be coated with some plastic material to prevent scoring in the event of contact resulting from the close clearance requirements between the shaft and housing. This concept shows some promise in turbomachinery applications; however, the problem of leakage prior to rotation and the possible instability of the interface rule out further consideration under this contract.

SEGMENTED CARBON SEAL

The face of the segmented carbon seal (Fig. 48) consists of a segmented carbon ring loaded radially against the seal housing ID. The nose of the seal is cut into one side of the ring. The segmented ring seals both against the rotating mating ring and the housing of the seal. A

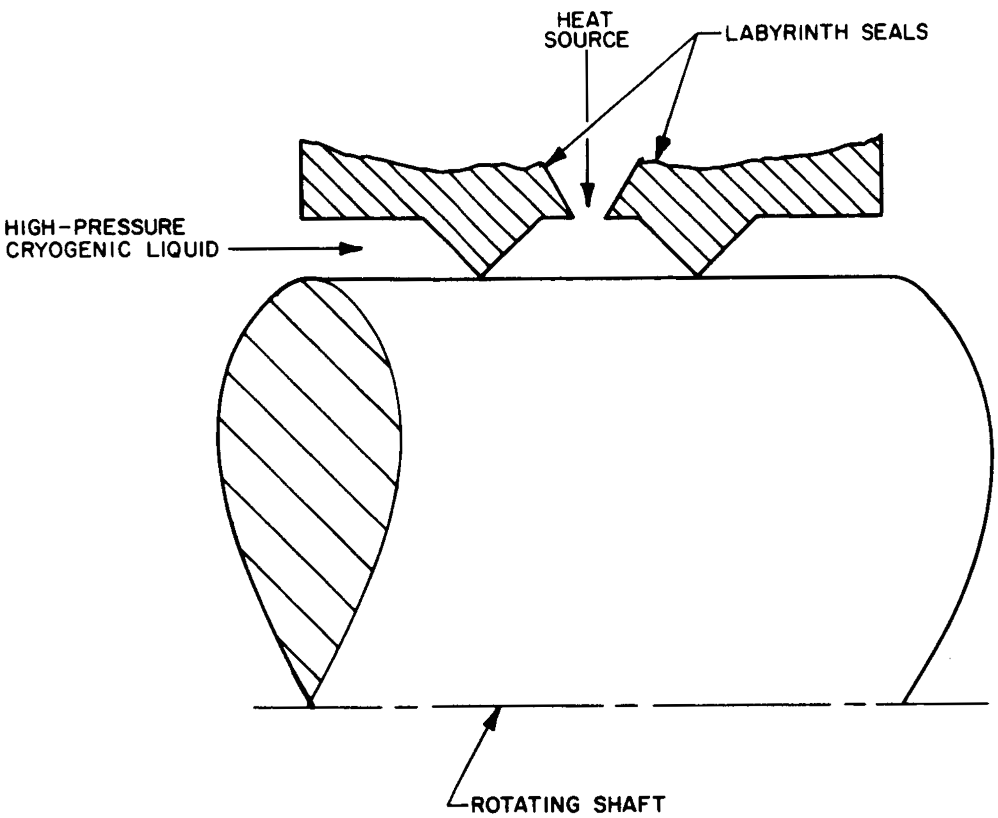


Figure 46. Heated Labyrinth Seal

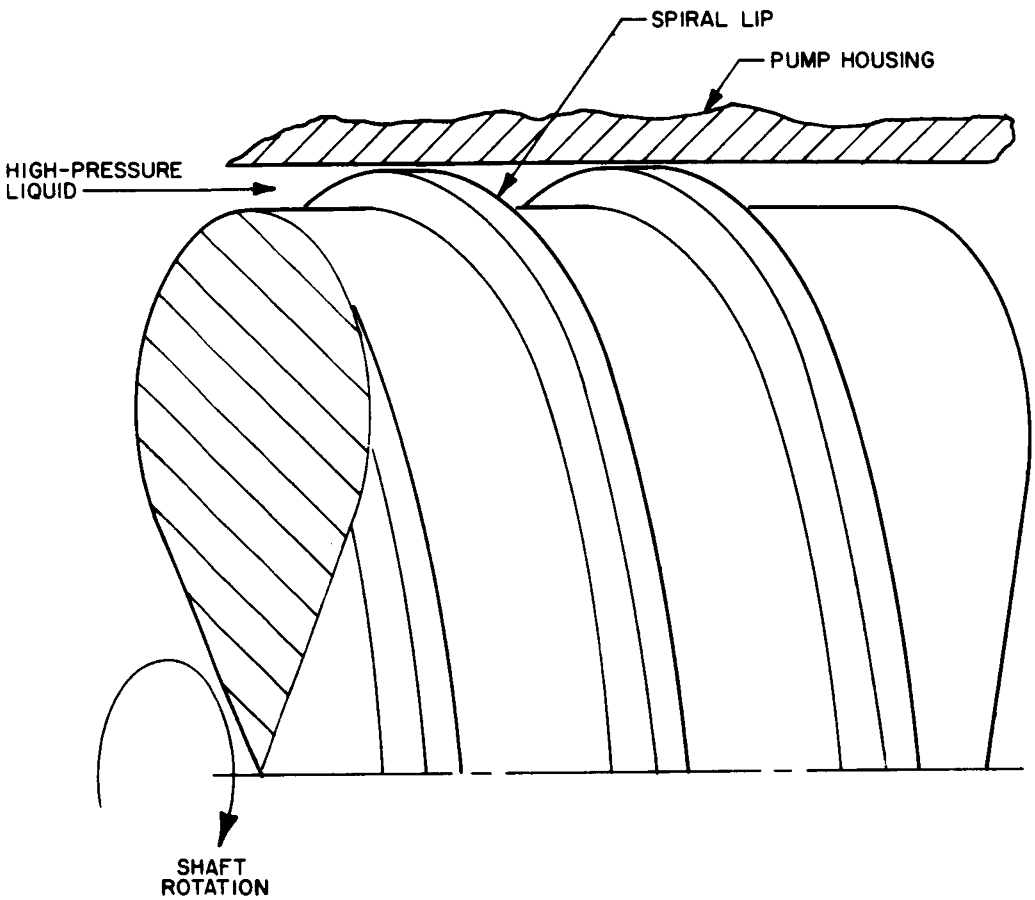


Figure 47. Shaft Pump Seal

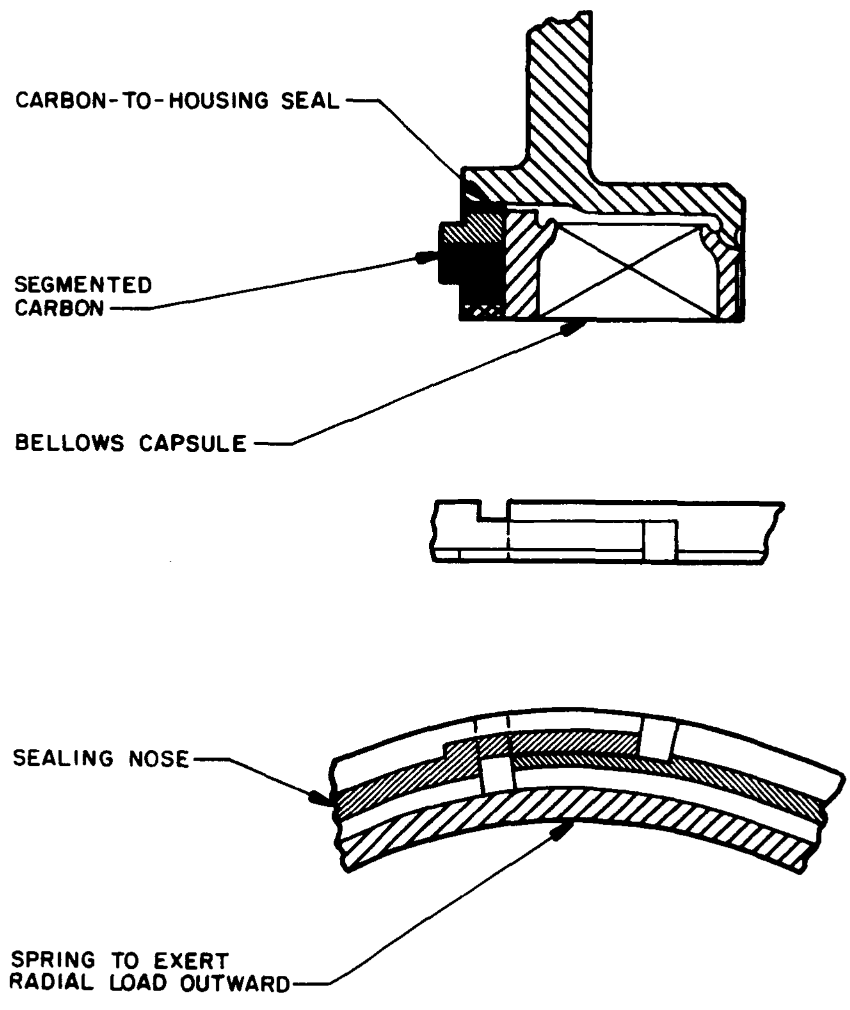


Figure 48. Segmented Carbon Bellows

bellows is used to load the carbon against the mating ring and also provide a second sealing element between the carbon and housing. This seal was eliminated from consideration because of excessive complexity.

BLADDER SEAL

The bladder seal configuration (Fig. 49) consists of a standard face seal rotating against a rotating mating ring, but incorporates a bladder or diaphragm-type secondary seal element (radial bellows). Analysis indicated that the radial bellows configuration would be excessively stiff axially with presently used seal construction materials required because of the high-pressure requirements and was eliminated from further consideration.

SEGMENTED SECONDARY SEAL

A standard composite design used in lip or elastomer-type seals would be used in conjunction with a segmented carbon secondary sealing element (Fig. 50). Because of the complexity of the design relative to its advantages further work was discontinued.

LABYRINTH BELLOWS SEAL

The labyrinth bellows seal (Fig.51) design consisted of a standard bellows-type seal to which was added a labyrinth-type seal between the seal housing and a cylindrical skirt attached to the seal carrier. The function of the labyrinth would be to damp pressure surges and thus increase the life of the bellows. The operation of this seal is quite similar to the piston damped seal which was selected for further analysis; therefore, further work on this seal was stopped.

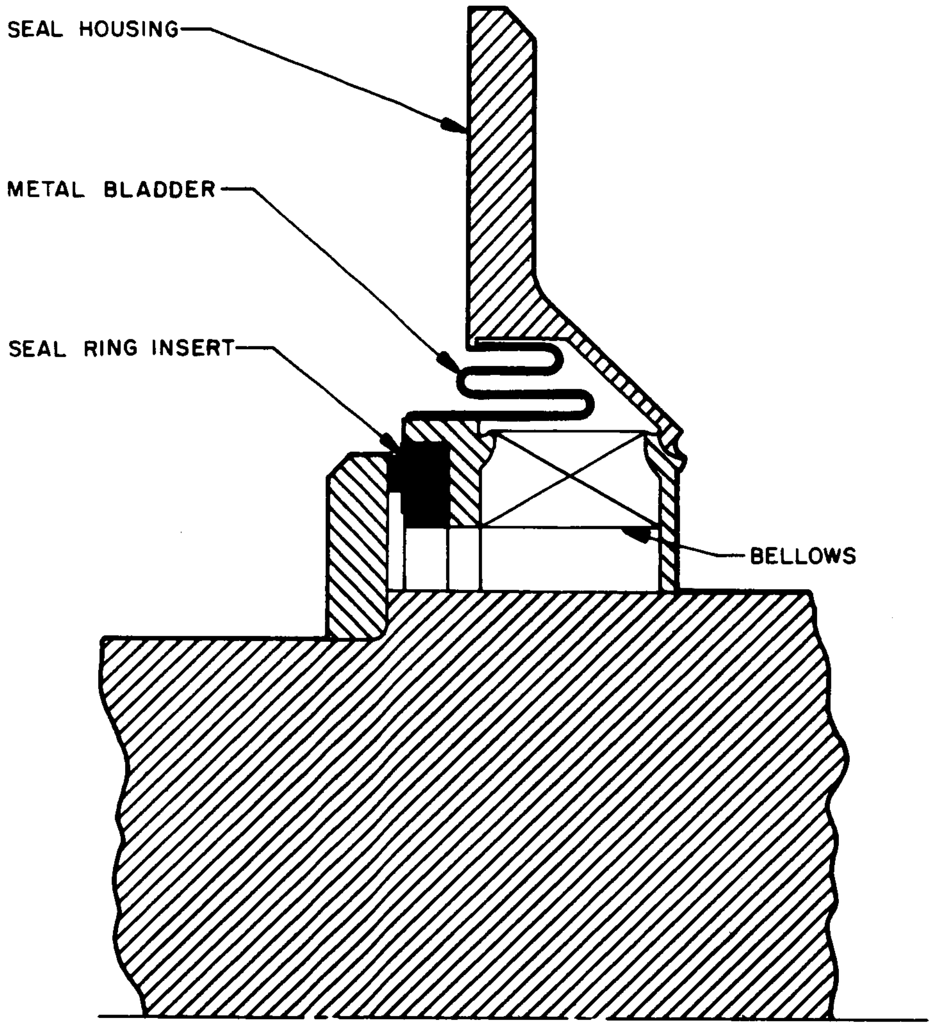


Figure 49. Bladder Seal

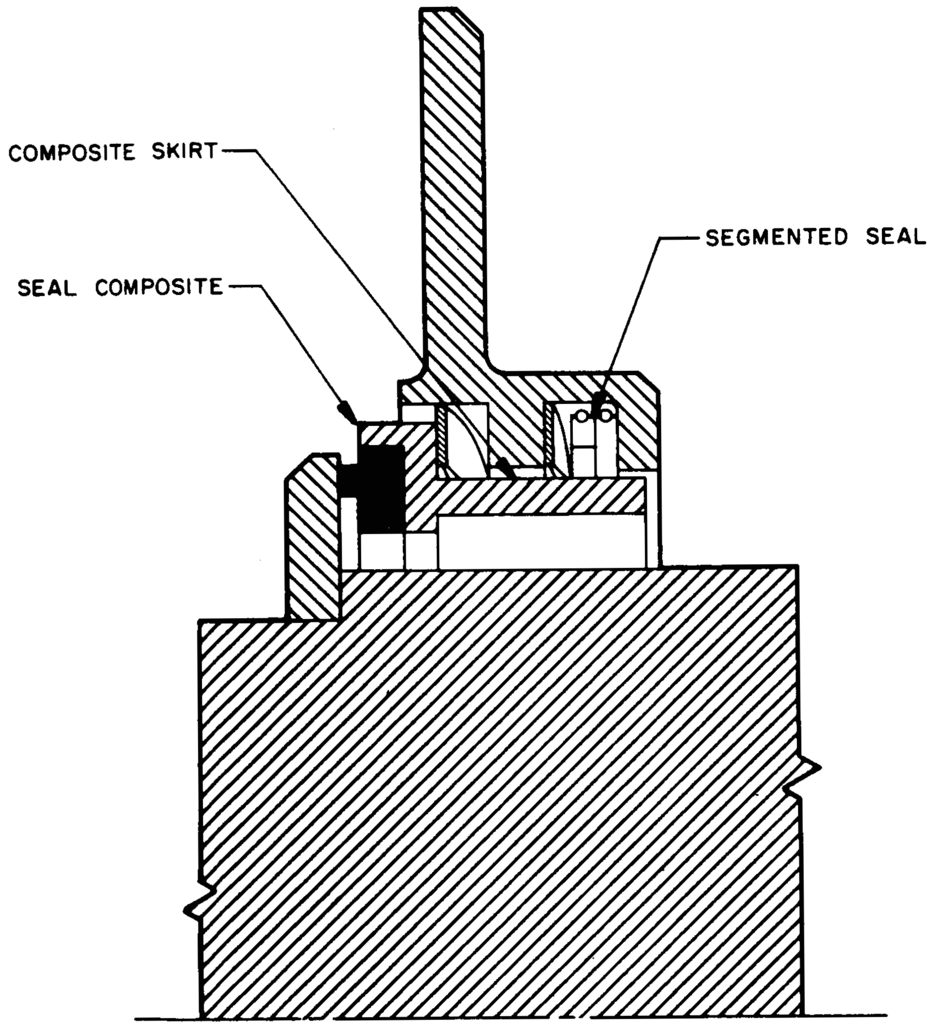


Figure 50. Segmented Secondary Seal

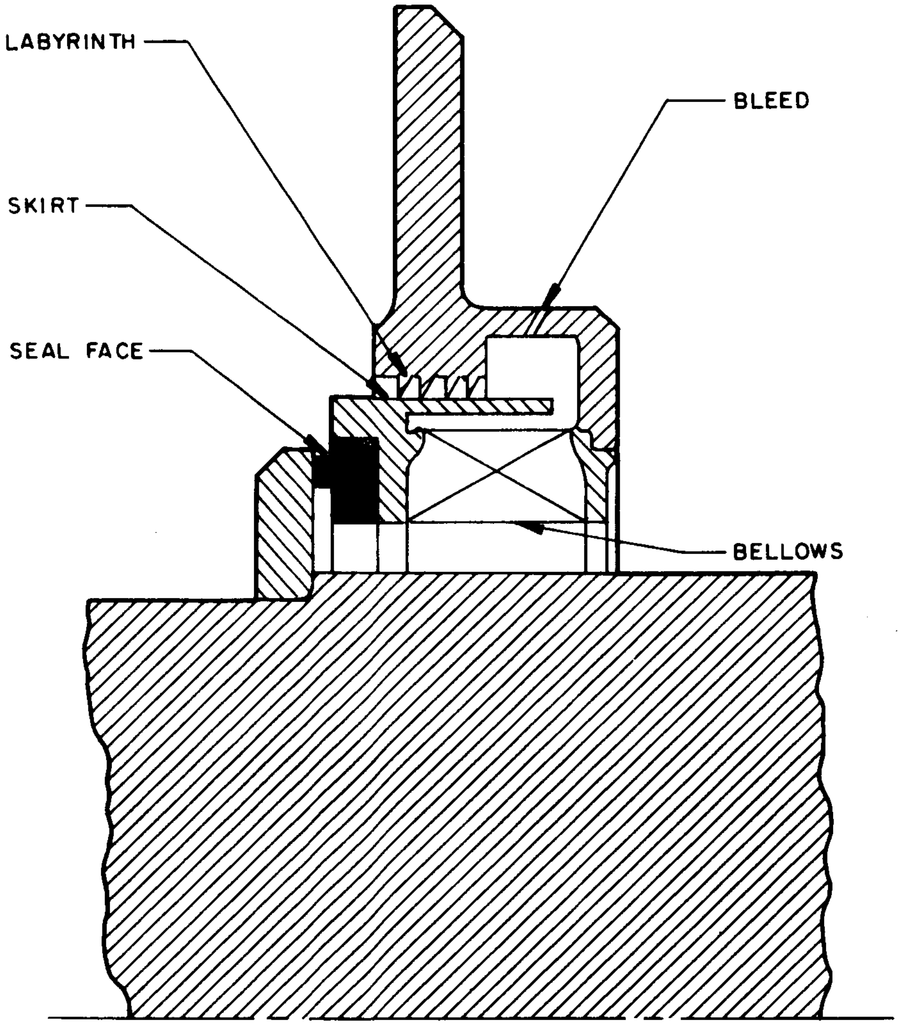


Figure 51. Labyrinth Bellows Seal

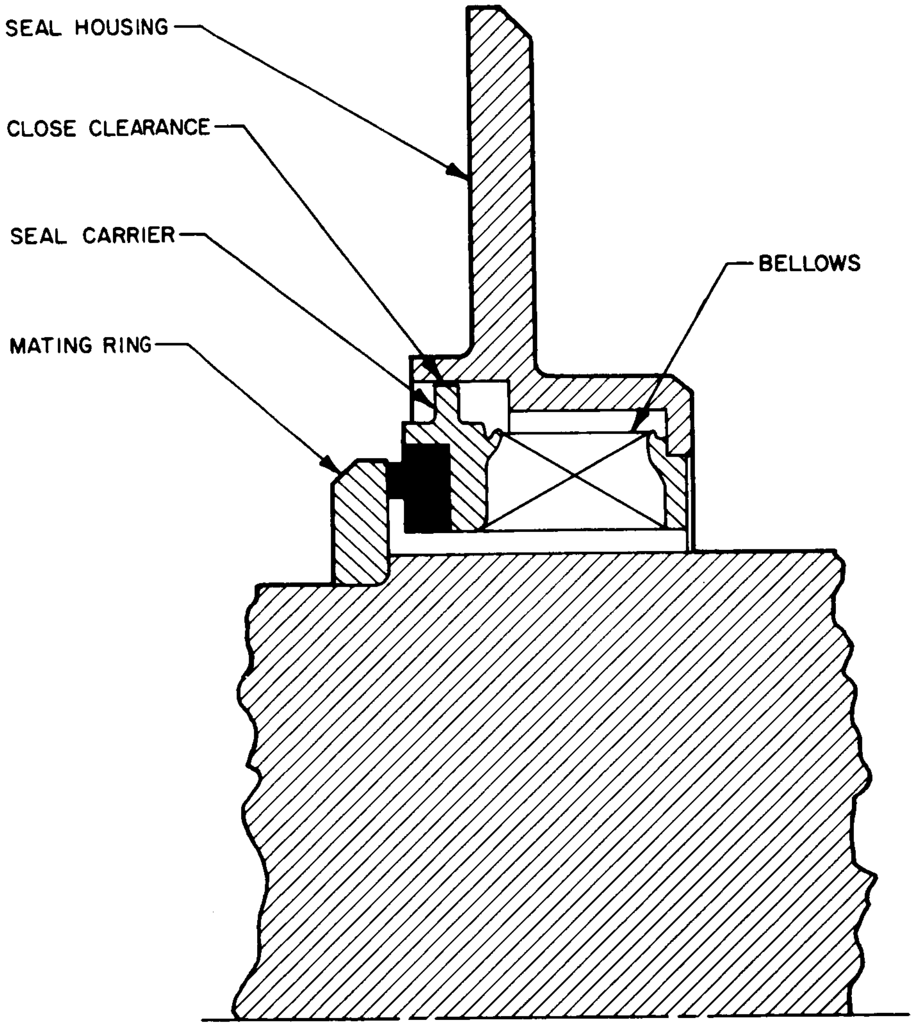


Figure 52. Viscous Damped (Piston Seal)

VISCOUS DAMPED SEAL (PISTON SEAL)

In the viscous damped seal (Fig. 52) the OD of the carbon carrier of a standard bellows seal was redesigned to incorporate a piston ring effect. The close clearance of the carrier to the housing allows a small flow area between the sealed cavity and the bellows cavity. Rapid movements of the seal face would be damped by viscous action of the fluid forced to flow through the small clearance area. This seal was selected for further analysis and testing.

PURGED DOUBLE LIP SEAL

The purged double lip seal design (Fig. 53) consists of two lip seals back-to-back, sealing against a skirt of a standard lip seal composite. Purge gas is introduced in the cavity between the lips and provides pressure to hold the lips against the skirt. The primary seal is a standard nosepiece and mating ring configuration. The double lip seal was selected for further analysis since it offered the advantages of (1) being adaptable to reactive fluids, and (2) having variable damping capabilities as the lip-to-skirt load can be controlled by the purge pressure.

PRESSURE-LOADED SEAL

The pressure-loaded seal (Fig. 54) consists of a close fitting cylindrical nosepiece fitted into an annular opening in the housing. High-pressure purge gas would be used to maintain propellant separation and also load the seal against the mating piece. To allow for misalignment,

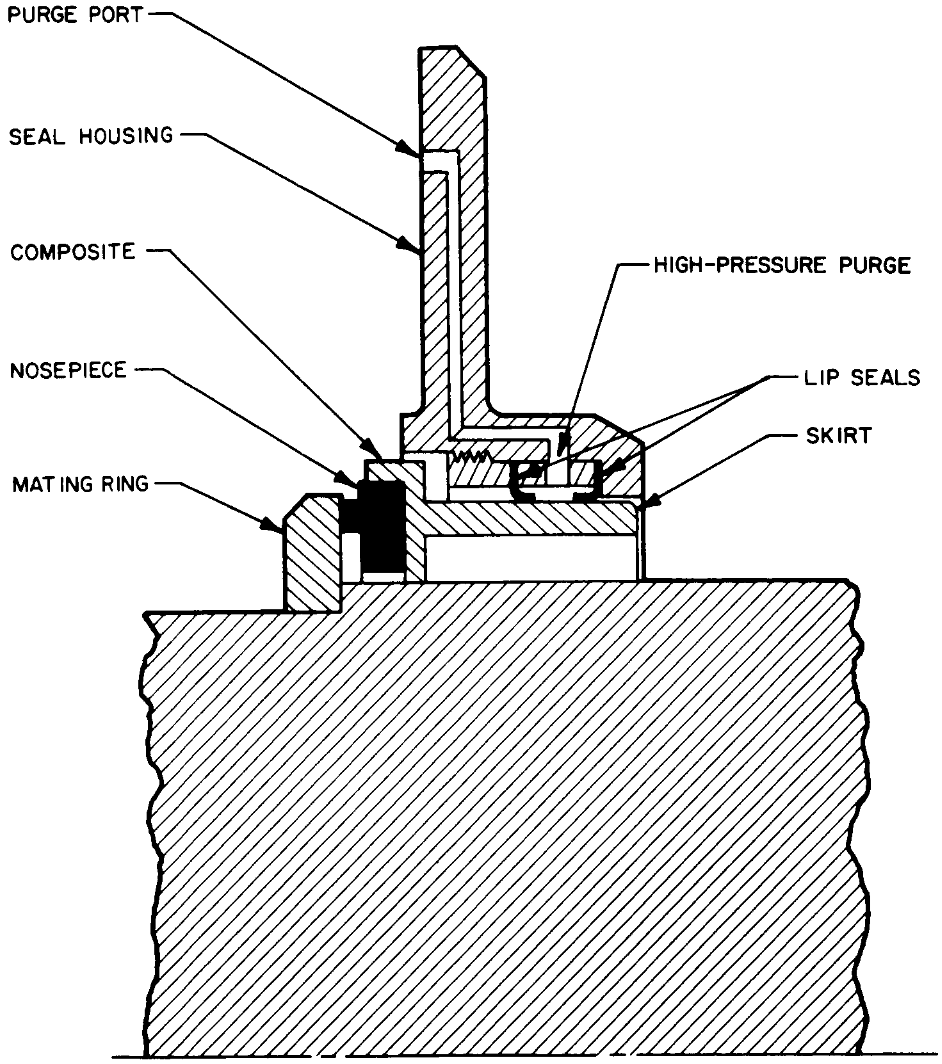


Figure 53. Purged Double Lip Seal

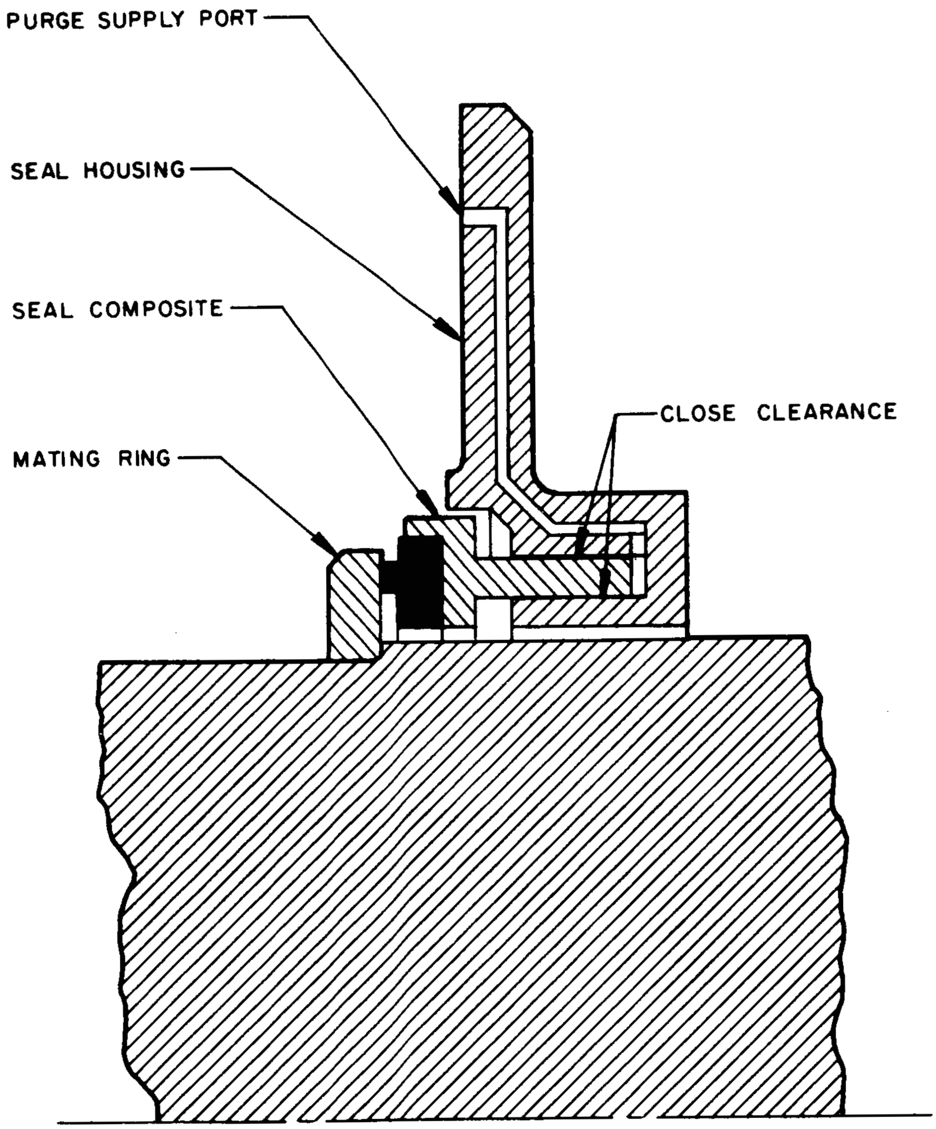


Figure 54. Pressure Loaded Seal

lips would be installed against the ID and OD of the nosepiece to allow clearance between the nosepiece and the housing. The low tolerance of this seal to misalignment resulted in its being dropped from further consideration.

VISCOUS DAMPED SEAL (ORIFICE SEAL)

Concentrically located bellows are joined to a common housing and a common carrier to form an annular cavity between the bellows. The area between the bellows is vented through an orifice in such a way as to provide viscous forces as the fluid is forced through the orifice. The orifice damped bellows seal (Fig. 55) was selected for further analysis and testing as it offers a feasible solution to increase the life of the bellows.

HYDROSTATIC SECONDARY SEAL

A hydrostatic-type journal bearing is located around the OD of the skirt. The composite skirt would be similar to that used on a standard lip seal. The hydrostatic portion would center the composite and the high-pressure gas would provide a backup seal. Two lip seals would be installed upstream of the hydrostatic bearing, one would seal the high-pressure propellant and the other would seal the high-pressure purge from the bearing. A drain would be provided between the lips.

The hydrostatic secondary seal (Fig. 56) is considered complex for the advantages offered and will have a low tolerance for misalignment. It will also require a high volume of purge fluid and, therefore, will not be considered further in this program.

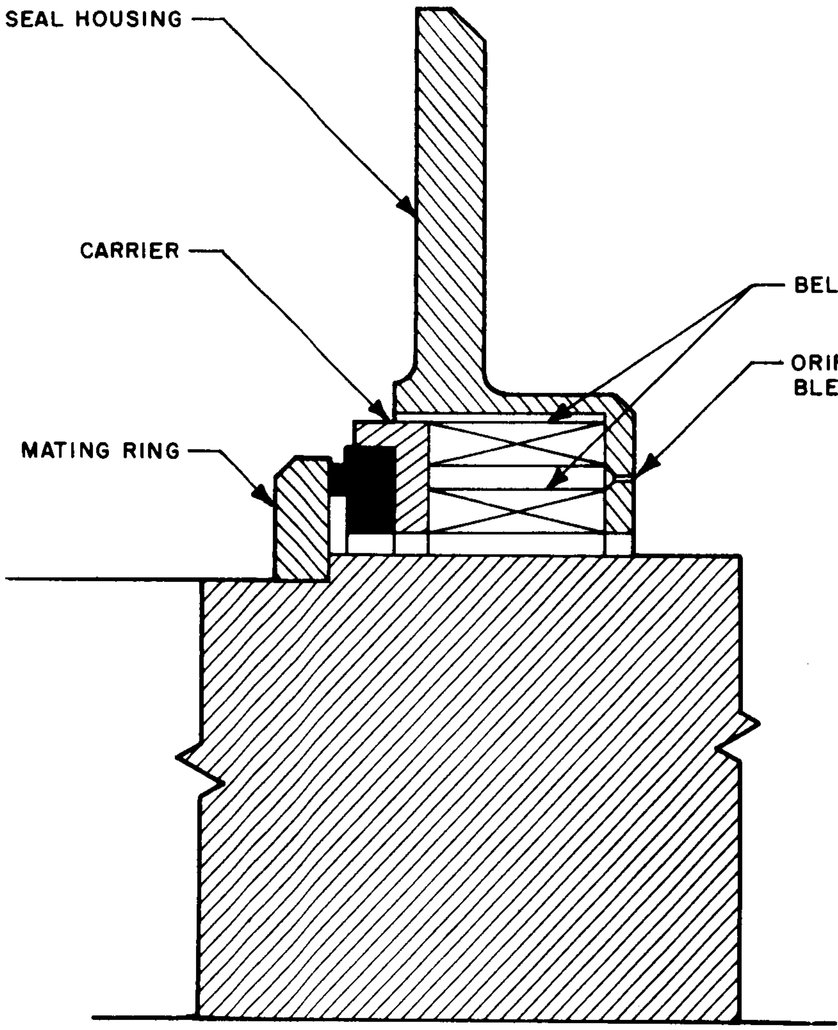


Figure 55. Viscous Damped (Bellows Seal)

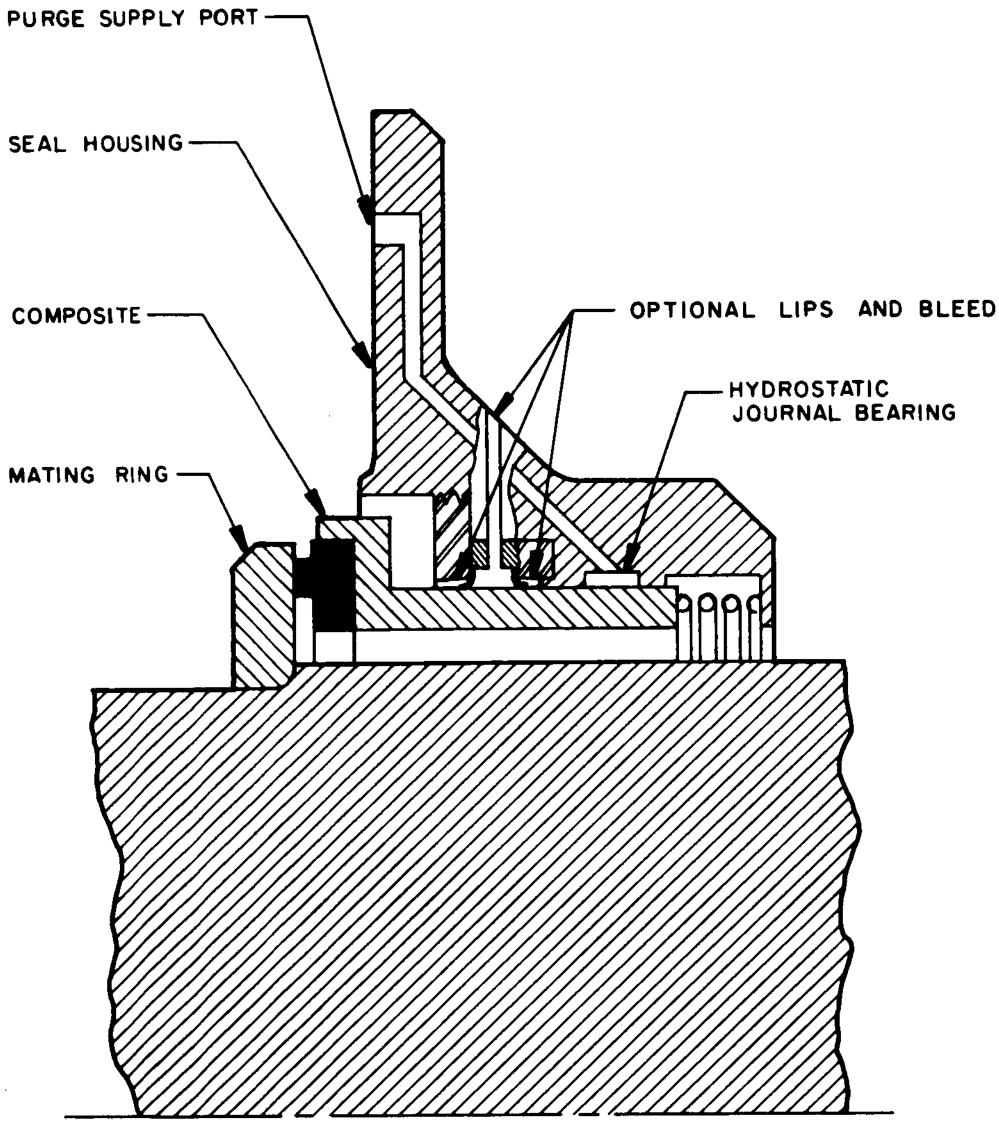


Figure 56. Hydrostatic Secondary Seal

PACKING GLAND SECONDARY SEAL

This seal (Fig. 57) utilizes a packing gland which would replace the usual lip or elastomer.

The limited applicability of the design using presently known materials led to rejection of this design for turbopump applications.

MACHINED BELLOWS SEALS

Naflex-type machined segments would be stacked together and welded to form a bellows assembly. Fig. 58 shows the configuration in which the inner surface of the segments are pressurized and Fig. 59 shows the segment reversed and the outer surface of the segments are pressurized.

The complexity of stress analysis and the estimated high cost of experimental development led to exclusion of this design from the present program. The configuration, however, is considered to have merit and could be analyzed further.

BALL CENTERED SEAL

The radial position of the secondary seal composite in this design would be maintained by a row of balls located between the housing and composite to obtain low frictional axial movement. A housing would be used behind the balls to provide sealing.

The ball centered seal composite (Fig. 60) while it would allow free axial movement would have little angular misalignment capacity; and, therefore, its applicability is limited. It will not be considered further for this program.

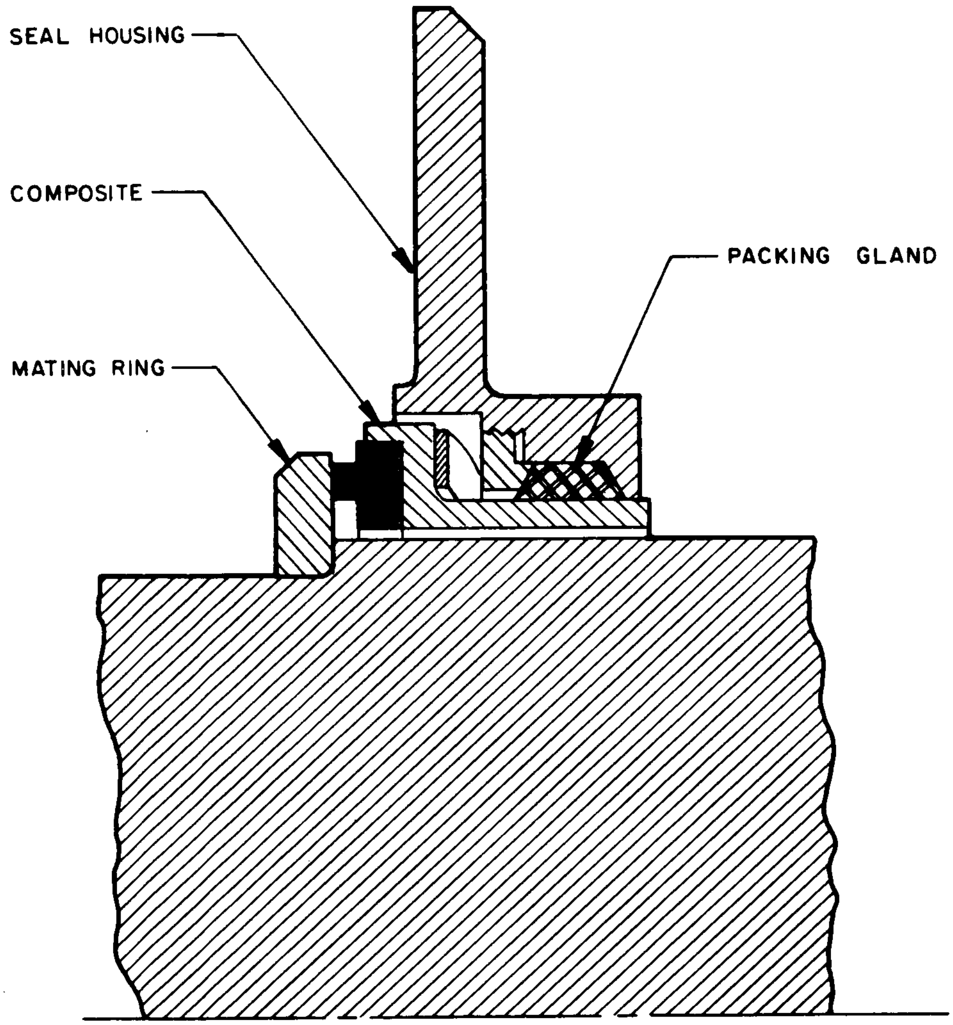


Figure 57. Packing Gland Secondary Seal

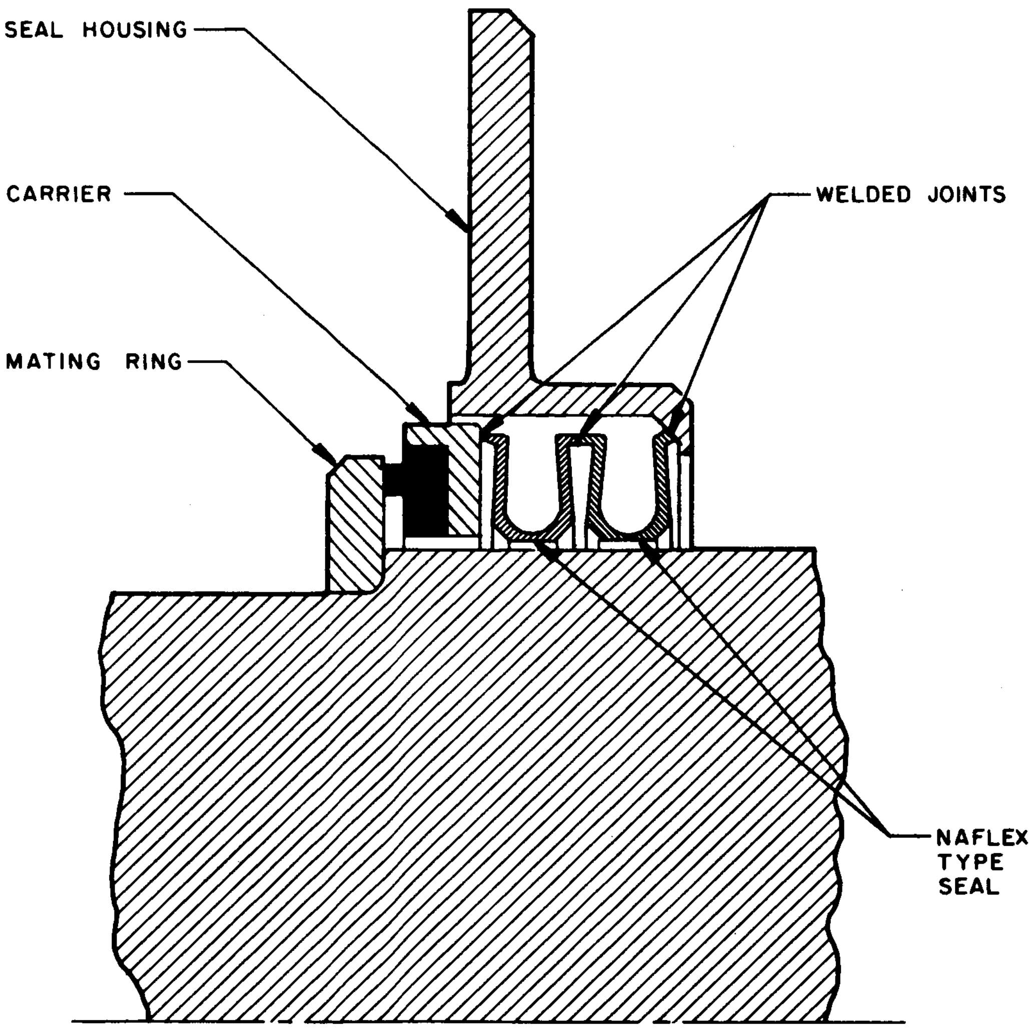


Figure 58. Machined Bellows (OD Pressurized) Seal

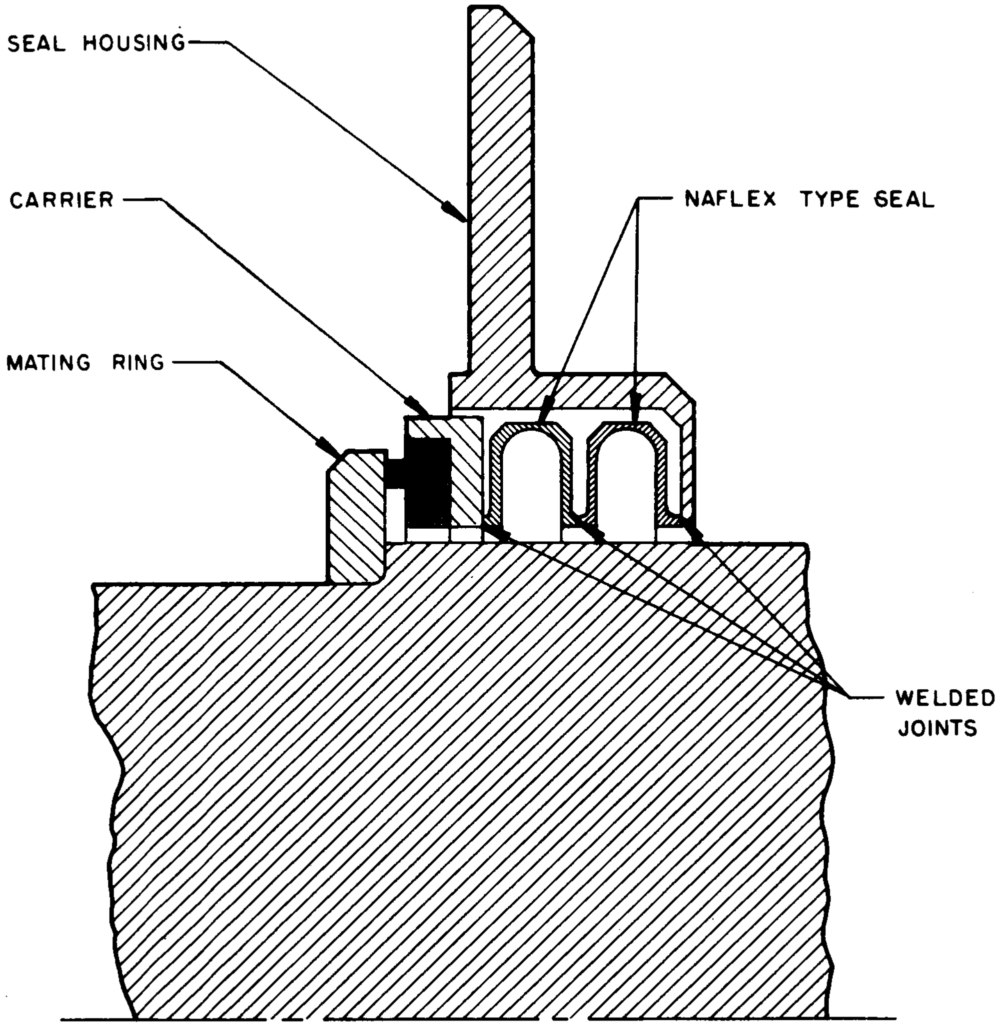


Figure 59. Machined Bellows (ID Pressurized) Seal

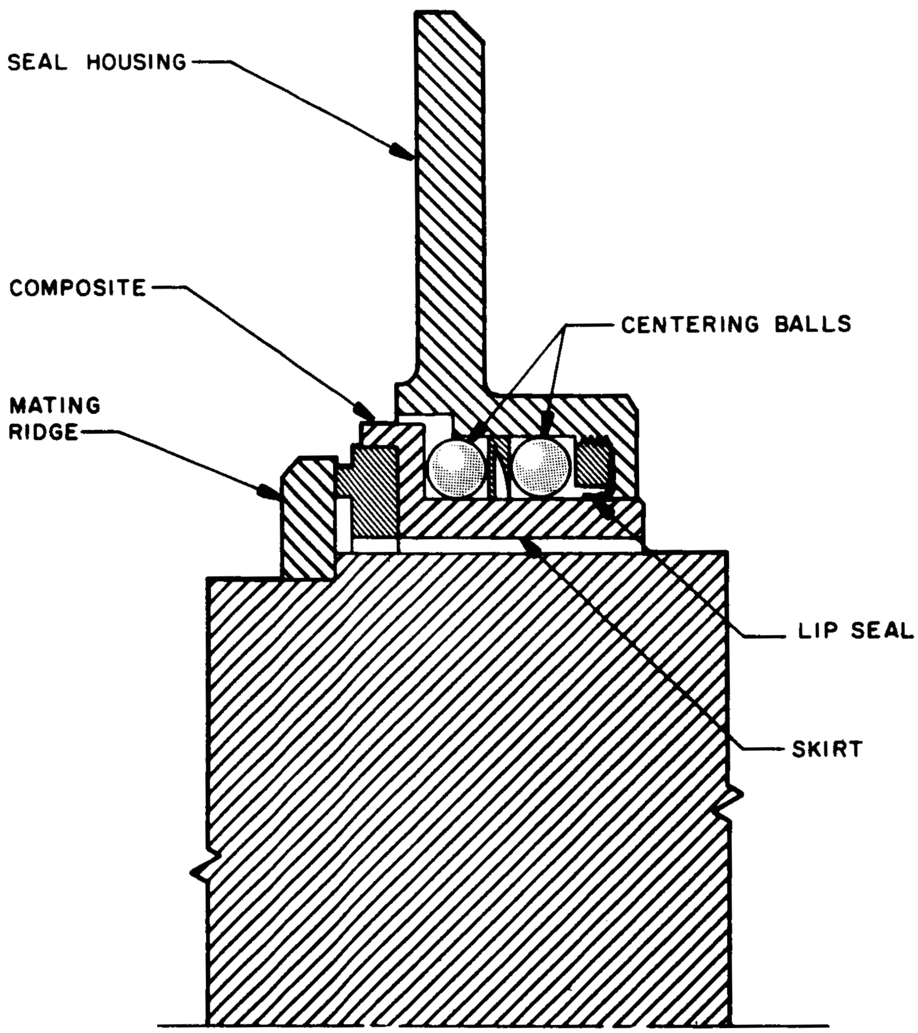


Figure 60. Ball Centered Seal

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13. ABSTRACT

A series of new type seal concepts were generated, and three of the most promising were detailed for fabrication and testing to evaluate the designs for future turbopump applications. Descriptions of the various concepts, basis for the final selections of the seals for evaluation, and results of testing are included.

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14. KEY WORDS	LINK A ROLE	LINK A WT	LINK B ROLE	LINK B WT	LINK C ROLE	LINK C WT
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Concept Design						
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